INDEXED

Marine Engineering

OCTOBER, 1905.

SEP 29 1905 PATENT OFFICE

THE SELF-DOCKING STEEL FLOATING DRY-DOCK DEWEY.

For the purpose of providing docking facilities in the Philippine Islands, which should be capable of caring for any ship of the United States Navy either now in those waters or to reach them in the future, the United States government, under an act of Congress approved July 1, 1902, called for proposals to be received by the Bureau of Yards and Docks in the Navy Department, and opened March 14, 1903. Under these proposals that of the Maryland Steel Company, of Sparrow's Point, Maryland, having been found to be the most satisfactory in every way, was adopted, and contract accordingly entered into with them April 20, 1903, for the construction and equipment of the dock. The contract was the more readily awarded to the Maryland Steel Company because of the fact that they had previously had a very successful experience in this line of work in the construction of the large dock owned by the United States government, and now in operation at Algiers, La., opposite the city of New Orleans.

FLOATING DOCK DEWEY RAISING ARMORED CRUISER COLORADO.

(Photograph by Groeninger, Baltimore.)

The specifications issued by the government contained the following general requirements:

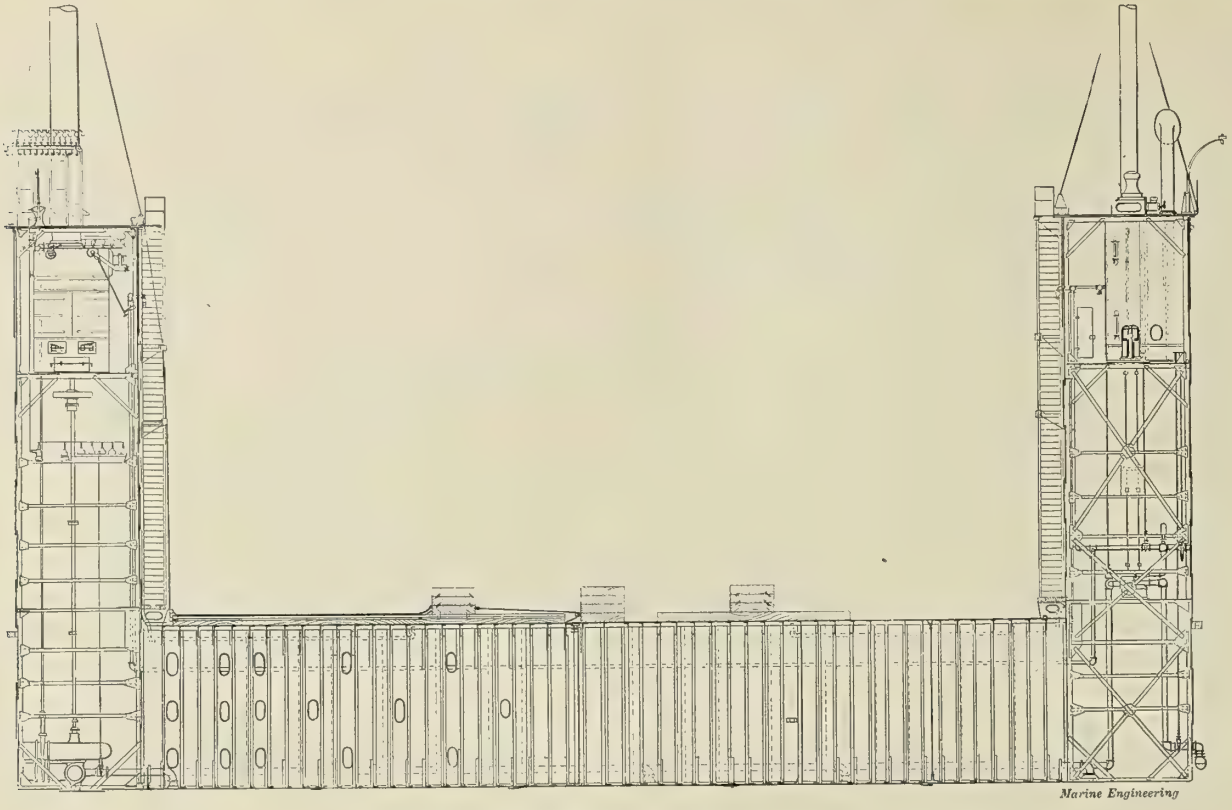
"The dock in general shall be an open-hearth steel structure, so designed and arranged as to be readily self-docking without the aid of divers or of auxiliary constructions. It shall be self-contained as to operating machinery, and capable of being towed safely from place to place without auxiliary bracing. It shall be of the general type, composed of watertight side walls, and body or pontoons with a general U-shaped cross section, and divided into a sufficient number of watertight compartments to give great stability, there being not less than six transversely.

"It shall be not less than 500 feet long over all outside of bracket platforms. It shall have a clear width between fenders of not less than 100 feet. The decks of the side walls shall have not less than 8 feet of clear height above the water with 30 feet draft over 4-foot keel blocks. The dock shall have a lifting capacity of not less than 16,000 gross tons uniformly distributed over its entire length, with the main deck not less than 2 feet above the water, and with not less than 1 foot of water in the compartments. It shall be designed to take all classes of vessels of the United States Navy, either centrally or with the center line of the keel 1 foot off the center line of the dock. It shall be so designed that the entire weight of a battleship may be safely carried by the main keel blocks, or one-half the weight on each line of docking keel blocks, and the side walls shall be designed to take shoring at any point which may be necessary.

"No portion of the dock or its connections shall have a stress of more than 10,000 pounds per square inch under the specified loads, nor more than 15,000 pounds per square inch in self-docking, with a wind pressure of 30 pounds per square foot of exposed surface.

"The working deck of the dock shall be flush plated and so

(Copyright, 1905, by Marine Engineering, Inc., New York.)



CROSS SECTION OF THE CENTRAL SECTION OF THE DOCK.

strengthened that docking keel blocks may be placed in any position, and the dock shall be so designed that the specified unit stress shall not be exceeded when the dock is pumped uniformly from all compartments to a freeboard of 2 feet. Under this latter condition the longitudinal and lateral deflection over the entire working deck of the dock shall not exceed 1 in 2,000. Within the limits of allowed deflection the ship-load shall be assumed to be perfectly flexible.

"The decks of the side walls shall have a clear passage fore and aft of not less than 5 feet in width, with a hand rail on the outboard side, and passage shall be provided from one side wall to the other, as well as telephone or speaking-tube communication between the two side walls, along the walls, and with the engine room.

"When self-docking, all under-water portions shall be raised to a clear height of not less than 5 feet for ready accessibility for inspection, painting, and repairs. The dock shall be operated by steam power and fitted with all the necessary boilers, engines, pumps, feed-water heaters, steam separators, and other auxiliaries desirable to make a first-class self-contained plant. This plant shall be of not less than 600 horsepower, with all boilers and engines suitably distributed to give the best results. A small

THE BATTLESHIP IOWA ENTERING THE SUBMERGED DOCK.

(Photograph by Groeninger.)

ARMORED CRUISER COLORADO ENTERING DOCK.

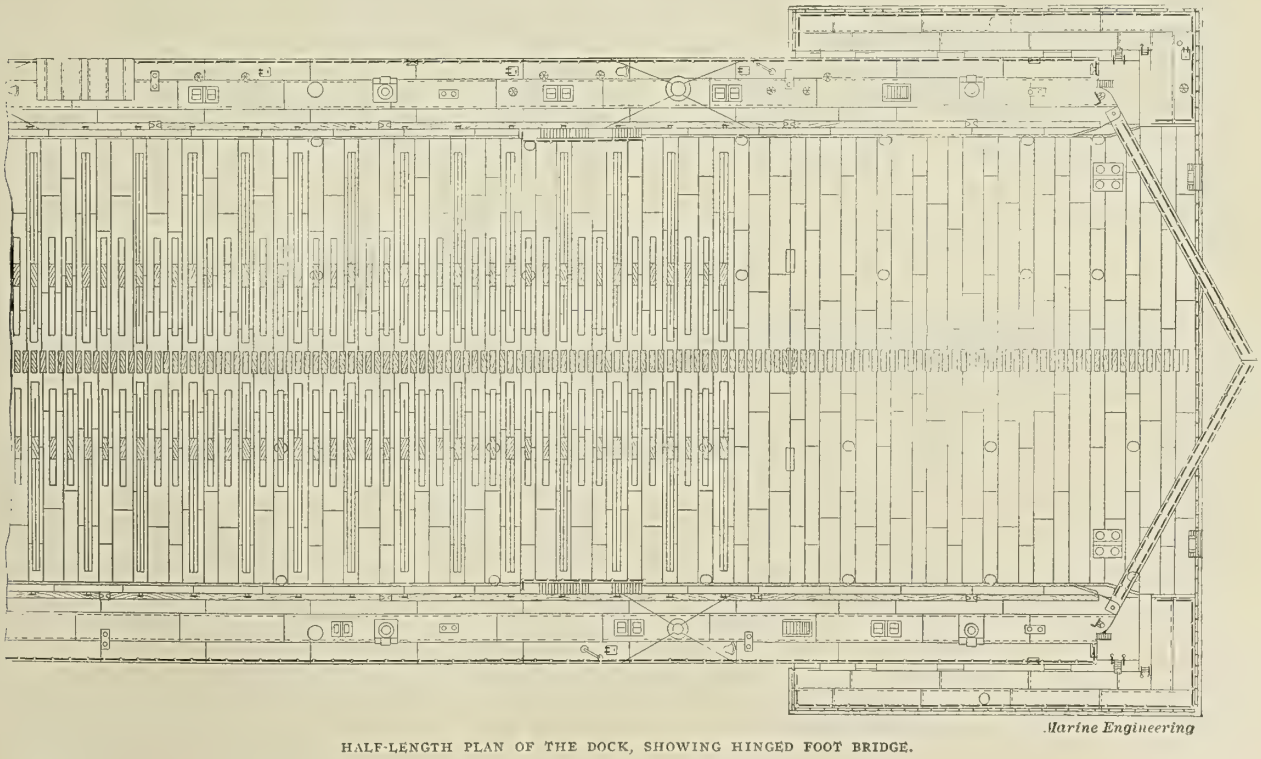
(Photograph by Groeninger.)

machine shop, suitable for light repairs to the dock, shall be installed in one side wall. Storage shall be provided for fuel and fresh water sufficient for two complete successive dockings of the maximum load. Such portions of the side walls above the engine decks as are not occupied by machinery shall be fitted as store rooms, and as quarters for officers and crew, with suitable mess arrangements. An electric-light plant shall be installed, and a blower system for ventilation of all working and storage places and quarters.

"The dock shall be designed to lift a load of 16,000 gross tons with a draft of 30 feet clear of the water in a period of four hours, the pumps being required to operate readily under a head of 35 feet."

The structural steel for the dock was required to have a maximum tensile strength of from 55,000 to 65,000 pounds per square inch, an elastic limit of not less than one-half the maximum tensile strength, and an elongation of not less than 23 percent in 8 inches. It was required to bend cold 180 degrees around a bar of diameter equal to thickness of plate, and to bend at a red heat 180 degrees flat, all without rupture on outside of bent portion. Similar requirements were made for rivet steel but with tensile strength of only 47,000 to 55,000 pounds per square inch, and for steel castings, whose strength should be upward of 60,000 pounds per square inch. The required tensile strength of wrought iron used was not less than 48,000 pounds per square inch, and all cast iron 18,000 pounds.

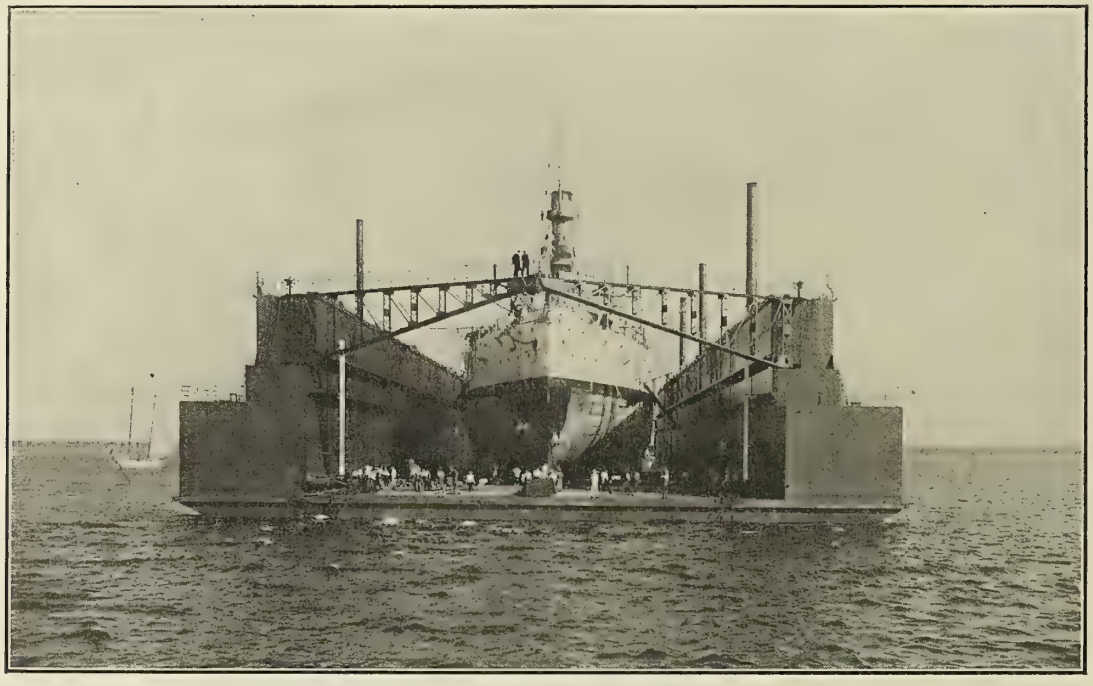
The design as worked out from these specifications gives a dock of a total length of 500 feet in three sections, of which the center has a length of 316 feet and the end sections each 90 feet, with a 2-foot open space between the center section and each of the others. The pontoons for the three sections are similar in transverse dimensions, the width being 134 feet over all and the



HALF-LENGTH PLAN OF THE DOCK, SHOWING HINGED FOOT BRIDGE.

depth 18 feet 6 inches throughout. This width of 134 feet is the extreme width over the side walls of the center section, whereas the end sections have a maximum width over side walls of 156 feet with free space between them 139 feet. This allows for the center section being raised by the two end sections and floated within their compass for any repairs or painting which might be found necessary, and this satisfies, so far as the center section is concerned, the requirements for self-docking. The end sections are docked by the center section by being turned through a horizontal angle of 90 degrees, and then floated in with their length of 90 feet between the side walls of the center section, which has a free space between fenders of 100 feet and a total space between walls of 106 feet. The length of the center section is seen to be sufficient to take in both end sections at once, with a clearance of 4 feet between them. The total height of the dock structure is 63 feet 8 inches, of which 45 feet 2 inches represents the height of the side walls above the main deck. The end pontoons are considerably lower than this, having a height above the dock of only 20 feet, which gives a total height for these sections of 38 feet 6 inches.

connected in the corners by the usual diagonal braces of 4½-inch by 3-inch by $\frac{7}{16}$-inch angle bars as well as with plate gussets. In the side walls the frames are built in this same manner, but are rendered more stiff by the introduction at intervals of horizontal struts. The transverse bulkheads are $\frac{7}{16}$ inch thick and are attached to both bottom plating and the deck by means of double angle bars and to the side walls by single angle bars. The central longitudinal bulkhead is of the same scantling as the transverse bulkhead and is fitted intercostally between them. The other longitudinal bulkheads are of the same scantling as the central and are stiffened by 7-inch by 14.75-pound channels spaced about 2 feet apart. All the plating throughout the dock is so fitted as to cross the frames at right angles. The bottom plating is, therefore, transverse between the side walls and longitudinal under them. Its thickness throughout is 24 pounds, except that under the side walls of the center pontoon, which is reduced to 22½ pounds. The middle strakes of the side-wall plating are 20 pounds thick, which is increased toward the bottom to 22½ pounds, and toward the top to 25 pounds. The upper two strakes in the center of the dock are doubled for the purpose of obtaining additional stiffness.



END VIEW OF THE DOCK, WITH BATTLESHIP IOWA ON THE BLOCKS.

(Photograph by Groeninger.)

As required by the specifications, the dock is divided throughout its length into six watertight compartments by five longitudinal bulkheads, of which one is located in the center, one on each side of the center and at a distance of 22 feet from it, and one immediately under the inside wall of the side walls of the dock. By means of transverse bulkheads numbering five in the center portions and two in each of the others the entire dock is divided into seventy-two separate watertight compartments, of which thirty-six are in the central section and eighteen in each end section. In addition to the transverse watertight bulkheads are other transverse bulkheads subdividing the dock into a cellular structure and spaced 8 feet apart fore and aft. These are all fitted with large manholes and limber holes to facilitate the free passage of water within the watertight cellular subdivisons.

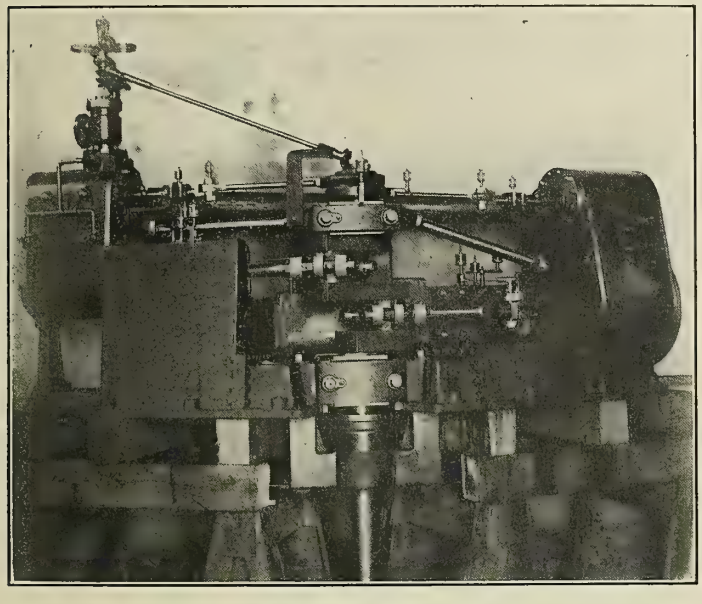
The frame spacing throughout the dock is 24 inches center to center, the frames consisting in 6-inch by 3½-inch by $\frac{7}{16}$-inch angles for the vertical and bottom members of longitudinal girders, while the top members consist in each case of channel bars which measure for the central frames 15 inches by 32 pounds and for the side frames 12 inches by 25 pounds. These members are all con-

The steam plant required by the government specifications is actuated by steam generated in three Babcock and Wilcox boilers of the marine type, each boiler having a grate area of 46 square feet with a heating surface of 1,750 square feet, thus giving a ratio of the latter to the former of 38 to 1. Steam is used mainly for the purpose of pumping, and for this there are three 24-inch centrifugal pumps and two 12-inch pumps of the same type. The latter are placed in the end pontoons and are used only for self-docking, while the larger pumps are used for the regular purpose for which the dock is designed. Each of the latter is operated by a horizontal compound engine having cylinders 14½ and 25 inches in diameter respectively with a stroke of 14 inches. The cranks are at an angle of 135 degrees. Each engine, with the regular steam pressure of 140 pounds per square inch, develops at a speed of 225 revolutions per minute about 225 indicated horsepower. The engines are placed far up in the wings and operate vertical shafts leading down to the pumps, which are very near the bottom of the side walls and which discharge outboard by means of horizontal pipes. The smaller pumps are each driven by simple engines with 12-inch cylinders and 10-inch stroke direct connected to

the pump shaft. The auxiliary machinery is operated by steam from a Babcock and Wilcox boiler with a grate surface of 18 square feet and heating surface of 750 square feet, and consists in the main of a 7-Kw. Sturtevant 110-volt generator driven by a simple direct-connected engine 6-inch by 6-inch, and a distilling apparatus with a daily capacity of 2,500 gallons.

The five pumps were furnished by the Morris Machine Works, of Baldwinsville, N. Y. They are placed in the bottom of the dock, with their suction openings directly on top of the main suction pipe running the full length of the dock; while the driving engines are placed on an upper deck on the side wall of the dock, and are directly connected to the pumps by a vertical shafting of 5-inch diameter. This is well shown in the illustration, which is one of the compound engines operating a 24-inch pump. The crank shaft is two-throw with the cranks opposite, and the cylinders at an angle of 135 degrees. The crank pins are 7½ inches in diameter and 7 inches long while the bearing section of the shaft has a diameter of 7 inches.

The 24-inch pumps are of the solid-shell type, and are made of cast iron of very heavy proportions, each one fitted with a bronze runner of the shrouded type 54 inches in diameter. As they are

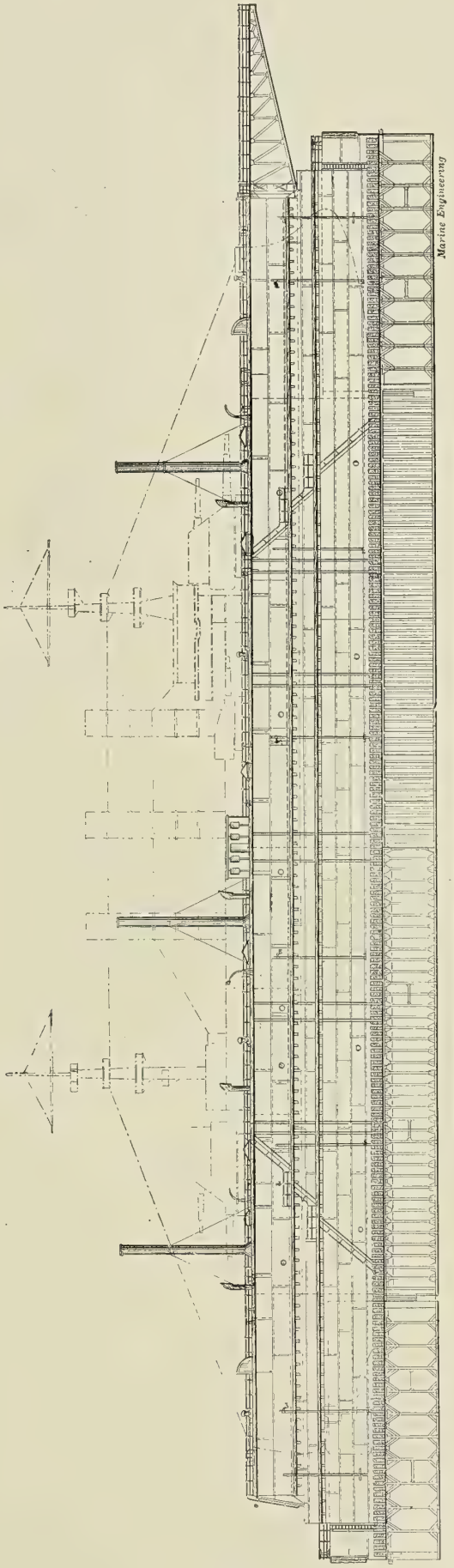


ONE OF THE PUMPING ENGINES.

worked on vertical shafts it is important that the thrust bearing be relieved in operation of as much pressure as possible. They have, therefore, been designed so that the water exerts an upward pressure on the runner or impeller. The weight of the crank shaft line shafting and impeller is carried by a ball thrust bearing hung directly under the engine framing. This is composed of two rows of 1¼-inch steel balls.

The 12-inch pumps are of the same general construction and have impellers of a diameter of 40 inches. Both types are fitted with throttling governors operated by gearing, so that in case the pumps should lose their priming the engines would be prevented from racing.

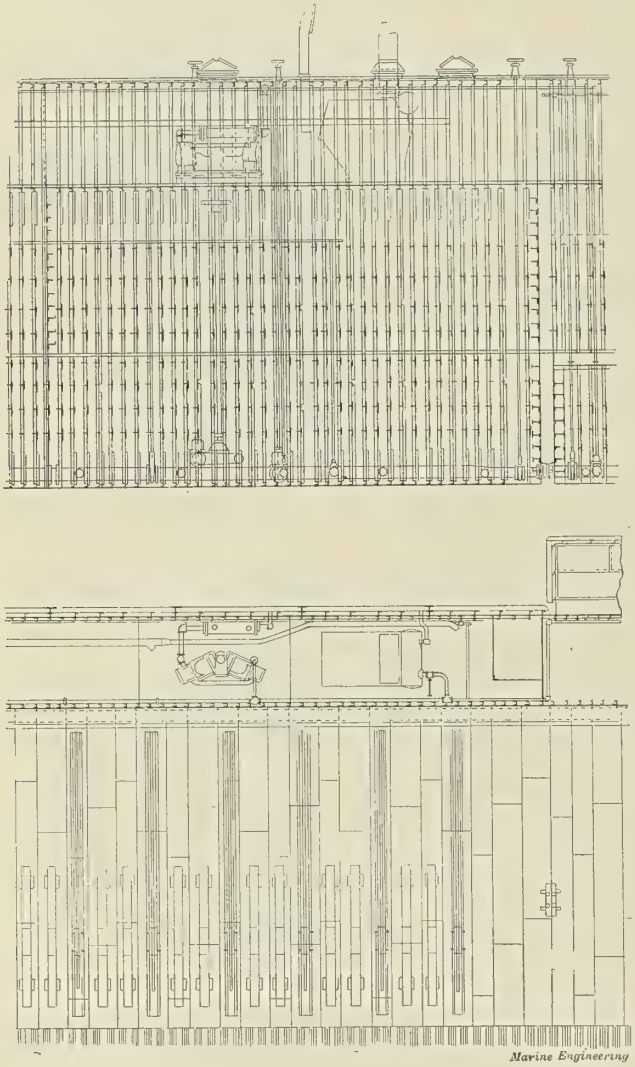
This dock is found by a comparison with existing structures to be considerably the largest of any dock yet built. The requirements, as per government specifications, call for a capacity to lift 16,000 tons with a freeboard of 2 feet. As an actual fact, it is found possible with this freeboard to lift 18,500 gross tons; and a little calculation establishes the fact that with a freeboard of 1 foot the lifting capacity of the dock is 20,400 tons, and, with the main deck awash, the lifting capacity is 22,300 tons. The comparison with other docks shows that the 18,500-ton lifting capacity with a freeboard of 2 feet is 1,000 tons greater than that of the Algiers dock built by the Maryland Steel Company, as before mentioned; and other large docks may be listed as follows:



INBOARD PROFILE OF THE DOCK, WITH A BATTLESHIP OF THE GEORGIA CLASS ON THE BLOCKS.

The Bermuda dock, built in England and now a part of the equipment of the English naval station in the Bermuda Islands, has a lifting capacity of 16,500 tons; the Austrian dock at Pola is rated at 15,000 tons; the German dock at Stettin has a capacity of 11,000; and the Pensacola dock, owned by the United States government, but which was formerly owned by Spain and was a part of the naval equipment at the port of Havana, Cuba, has a capacity of 10,000 tons.

The *Dewey*, according to the original contract, was to have cost $1,124,000, but certain modifications and additions have crept in since the work was started, and the ultimate cost will somewhat exceed this figure. The steel used in construction amounts to about 11,000 tons, which is held in place by more than two million rivets. This great weight, added to the lifting capacity of the



INBOARD PROFILE AND HALF-PLAN, SHOWING PUMPING MACHINERY.

dock, will represent a total displacement of about 30,000 tons at a draft of 16 feet 6 inches, but it is plain that for the dock to exercise its legitimate functions, a depth of water of above 50 feet will be required in order that a ship drawing 30 feet of water can float in over the keel blocks while the dock is submerged. At light draft the dock draws only 6½ feet.

The accommodations on the dock are designed for nine officers and twelve men, with state rooms and mess hall, a well-appointed galley, and a finely-equipped library.

The dock was constructed under the immediate supervision of Mr. Leonard M. Cox, Civil Engineer, U. S. N., working in conjunction with Mr. S. Anderson, Superintendent of the Dock Department, Maryland Steel Company.

The first test of the dock was made by means of the armored cruiser *Colorado*, which had at the time a displacement of 13,300 tons distributed over a length of about 500 feet. The ship was lifted clear of the water in 2 hours 16 minutes. A more severe test was that connected with the battleship *Iowa*, which had a displacement of 11,600 tons and a length on the keel blocks of about 330 feet. This ship was lifted clear of the water in 1 hour 37 minutes, but the pumping was continued until the dock had such a freeboard as to represent the raising of a 16,000-ton ship, which was accomplished in 2 hours 42 minutes altogether. In this latter case a deflection in the length of the dock was found amounting to about 2 inches, which had increased at the expiration of 48 hours to 4 inches, but which in the bearing length of the ship did not exceed 1¾ inches.

The self-docking test occupied about fifteen days, but it is expected that this time will ultimately be reduced to eight or ten days, which will be a record for this operation.

The dock is to be turned over to the Navy Department about the first of October, and arrangements will then be made for towing it to Olongapo, its ultimate destination. It is expected that the process of towing the dock this distance of some 13,000 miles will require about five months.

NOTES ON THE CALCULATION OF SIZE OF RUDDER STOCKS.

BY J. W. CLARY.

Empirical formulæ for calculating the diameter of the rudder stock of a vessel are given by most of the classification societies, and in the various shipbuilders' handbooks and text books of naval architecture. These formulæ show great differences in results, and in some cases, notably those of the formulæ given by the various classification societies, are applicable only to the form of rudder usually found on merchant vessels. In these notes an endeavor has been made to state the theory of these calculations, to give the results of tests from ships in service, and to derive formulæ which may be quickly applied and are sufficiently flexible to suit any case. In this connection the following authorities have been consulted:

Pollard et Dudebout—Theorie du Navire.

William F. Durand—Paper read at the seventh general meeting of the Society of Naval Architects and Marine Engineers, on "The Action of the Rudder."

L. Callou—Cours de Construction de Navire, 1902.

A. Croneau—Construction Practique du Navire de Guerre, 1894.

E. Gayde—Cours Practique de Construction Navale, 1895.

Paper read by Arthur J. Maginnis before Institute of Naval Architects, April 16, 1886, on "A New System of Steering Gear, and Rudder Strains Recorded by It."

And "Note au sujet du calcul du Moment Resistant des Gouvernails par M. de Courville," Bulletin de l'Association Technique Maritime, No. 7, p. 57.

It seems to be agreed that the best method of estimating the stress in a rudder stock and other fittings connected with the rudder is to calculate the pressure on the rudder blade and its moment about the axis of the stock by means of formulæ derived by Joëssel for the general action of water on planes moving obliquely to their normal, modified by a *coefficient of reduction* obtained by experiments on vessels under actual conditions of service. The number of these experiments is limited, the types of vessel on which they have been made differ widely and the speeds at which the experiments were made were, in the majority of cases, low, so that no actual figures for large, high-speed vessels are available; but from the data which exists coefficients of reduction can be taken which may be used with judgment.

It is true that the stresses on the rudder stock due to the shock of waves may be five or six times as great as the stresses due to putting the helm over in still water (S. S. *British Prince*—maximum pressure on gear to put rudder hard over 5.6 tons at 13

knots speed in calm, while springs were broken in indicators during a storm, showing at least 30 tons pressure) and that it is impossible to calculate these dynamic stresses; but in the majority of calculations of the strength of structures or members of a structure that are to support live loads or undergo any dynamic stresses, the calculations must be for statical stresses and a factor of safety applied to cover any stress that is likely to occur. Even if it is considered best to establish the size of the rudder stock entirely from the experience of the designer, a more or less standard form of calculation would serve as a basis of comparison (as is the case in calculations of statical stability and of strength of ships) and would be of value.

The theory of these calculations is well expressed in Theorie du Navire (Pollard & Dudebout), Tome IV., page 26, *et seq.*, a translation of which follows:

"To estimate approximately the resistance of a rectangular rudder with a vertical stock the formulæ for a thin plate, found by M. Joëssel, may be used, assuming that, in the relative movement of the ship and the water, the stream lines have a constant speed in relation to the rudder blade equal to the initial speed of the ship and parallel to its length. . . .

"But the hypothesis that has been made, that the direction and the velocity of the stream lines remain constant, limits the application of these formulæ to the almost useless case when a ship remains fixed, moored in a current with a velocity V_0, parallel to its keel and coming from ahead. But here would be the place to remark that the angle of incidence of the stream lines on the rudder blade is not equal to the helm angle α, but greater than this angle by a quantity which depends on the mean angle of the after part of the water-lines of the hull for the height of the rudder. Nevertheless, this influence of the form of the hull on the stream lines which strike the rudder blade can be neglected because this same form favors the return, in a horizontal direction, of the water to the back of the rudder blade and thus restores, more or less completely, on the back the excess of resistance which it has caused on the front. It is, besides, necessary to consider, especially if the stern is cut squarely as in the case of a barge, that the stream lines acting on the rudder would have their relative velocity diminished by the partial influence of the liquid mass.

"In reality, the rudder is not made for action on a ship moored in a current, but on a free ship, and then it causes a horizontal movement of the ship in a curved line. . . . Now this movement reacts on the action of the rudder and modifies its intensity by changing the direction of the stream lines which are about to strike the rudder blade. In other words, the above formulæ for the direct resistance and for the moment tending to return the rudder to a fore-and-aft position . . . are applicable only for a very short interval of time at the beginning where these quantities are still, if not very small, at least far from their maximum values, which alone interest us for the judicious determination of the dimensions of the various pieces used to swing the rudder blade (rudder stock, tiller, tiller ropes, wheels, mechanical apparatus, etc.).

"To solve the problem and to find, in the usual case of the free ship, the maximum reacting moment when the helm angle has reached the final value $\alpha = \alpha_1$, that it is desired to keep constant, it is necessary to study first the movement of the ship during the 'maneuvering period,' or, in other words, during the period while the rudder blade is swinging.

"We shall first notice that the formulæ (of M. Joëssel) applied for the definite angle α_1 will give too large results for the two following reasons:

"1. Admitting that at the end of the time used in swinging the rudder blade the ship would not yet have turned horizontally, the final speed V_1 in the direction of the keel would be less than the initial speed V_0 on account of the component of the additional resistance (of the rudder) in the direction of the keel.

"2. At the end of this period the ship has already acquired an angular velocity at right angles to the keel, which diminishes the angle of incidence of the stream lines on the rudder blade by a certain angle ε, and consequently the final moment tending to return the rudder to a fore-and-aft plane.

"It follows from this that the ratio of the actual final moment tending to return the rudder to a fore-and-aft plane to the moment given by M. Joëssel's formulæ for $\alpha = \alpha_1$, is equal to:

$$\frac{V_1^2}{V_0^2} \times \frac{\sin(\alpha_1 - \varepsilon)}{\sin \alpha_1} = Q.$$

This is the *coefficient of reduction* to be calculated, with which it is necessary to modify the results given by the formulæ if it is desired to know the actual final reacting moment, and the work which the helmsman or the steam engine must furnish to swing the rudder blade."

Following out this line of reasoning, Prof. Durand (Trans. Soc. Nav. Arch., Vol. VII, page 267, *et seq.*) takes the case of a vessel of 8,000 tons displacement, 400 feet long, 50 feet beam, 22 feet draft, area of rudder 200 square feet, extreme helm angle 35 degrees, time required to put helm over 10 seconds, speed 25 feet per second, or about 14 knots; and, by methods of his own for solving differential equations, calculates the value of the helm moment, showing the reduction, as compared with the value obtained by Joëssel's formulæ, due to the decrease in the angle of incidence of the water striking the rudder and in the speed; giving in this case a value of .64 for the coefficient of reduction. He also constructs various curves, of which one in particular shows the angle by which the effective angle of obliquity of the flow of water to the rudder (denoted in the foregoing translation by ε) is reduced, which, for the assumed case, amounts to 12.5 degrees at a maximum angle of 35 degrees.

Joëssel's formulæ for the pressure and its point of application on a rectangular plate submerged in a current are as follows:

$$P = 1.622 \frac{\sin \alpha}{.39 + .61 \sin \alpha} \cdot \frac{w}{2g} \cdot AV^2 \quad \text{(a)}$$

$$x = (.195 + .305 \sin \alpha)\, b \quad \text{(b)}$$

Where P = normal pressure on the plate in pounds.
A = area of plate in square feet.
V = velocity of current in feet per second.
α = inclination of face of plate to direction of current.
w = weight per cubic foot of water in pounds.
g = acceleration due to gravity in feet.
b = breadth of plate.
x = distance of center of pressure from forward edge of plate.

For V = speed in knots; $w = 64$ pounds, and $g = 32.16$ feet.

$$P = 4.6\, AV^2 \frac{\sin \alpha}{.39 + .61 \sin \alpha} \quad \text{(c)}$$

For $\alpha = 35°$ (usual maximum helm angle) $\sin \alpha = .5736$.

$$P = 3.57\, AV^2 \quad \text{(d)}$$

$$x = .37\, b \quad \text{(e)}$$

The following table of coefficients of reduction, to be applied to the foregoing formulæ, obtained by experiment on vessels under service conditions, is taken from Prof. Durand's article (Trans. Amer. Soc. Nav. Arch., Vol. VII., page 279), to which have been added the results of trials on the *Condor* and *Epervier*, which may be found in Theorie du Navire (Pollard and Dudebout), Tome IV., page 36.

It will be seen that these coefficients (which represent the ratio of the actual twisting moment on the rudder stock to the moment as estimated by the formulæ) vary from .35 to .80 under conditions which are well stated in Theorie du Navire, Tome IV., page 37, discussing the results for the *Condor* and *Epervier:*

"The above table shows that for a given ship and for a given speed of maneuver (of the helm):

"1. When the time required to put the helm over remains constant the coefficient diminishes in proportion as the initial speed in a straight line increases, which agrees with the theoretical

deductions. In other words, the actual final reacting moment increases much less rapidly than the square of the initial speed.

"2. When the initial speed in a straight line remains constant the coefficient decreases in proportion as the time required to put the helm over increases, which also agrees with the theoretical indications, although these indications have been found for a very small length of time and for a constant helm angle. . . .

"It is clear that for the same angular velocity of helm about the rudder stock, a ship of large mass, and, at the same time, very long and very fine at the ends, presenting, in a word, a great resistance to turning in relation to the power of its rudder, would lose, during a given time of maneuver of the helm, a smaller fraction of its initial speed, and would also acquire, during this time, a smaller drift angle, so that its coefficient of reduction would be greater than the above (for the *Condor* and *Epervier*), and its actual final reacting moment would approach more nearly the results given by the formulæ of M. Joëssel. Experiments have shown this, for dynamometric experiments made during turning trials of the *Amiral Baudin* by M. l'Ingenieur de la Marine de Courville have given for this ship, at an initial speed of 14 knots, coefficients of reduction amounting to 1.53 to 1.60 times those of the *Condor*."

Or, in other words, a small coefficient of reduction is associated with small vessels, relatively large rudder areas, high speed of ship, and low speed of helm, and a large coefficient is associated with large vessels, small rudder areas, low speed of ship, and high speed of helm.

Professor Durand says: "In the French naval designs the value .48 has been very commonly taken," and this is value corresponding to the constant given by Croneau for ships of 18 to 20 knots speed (Construction practique du Navire de Guerre, Tome second, page 455).

Callou, in his "Cours de Construction du Navire," of 1902, Tome second, page 125, says:

"According to actual practice the pieces for swinging the rudder are placed in such a way as to permit a maximum angle of inclination of between 30 and 35 degrees; sometimes, because of difficulties of installation, this is decreased to 28 degrees. This accomplished, the maximum pressure sustained by the rudder blade is calculated by means of the following formula:

$$P = \frac{KSV^2 \sin \alpha}{.195 + .305 \sin \alpha},$$

in which is substituted for V the value of the maximum designed speed of the ship, for α the maximum value permitted by the installation of the members controlling the rudder, and for K a value equal to 20 kilos, at least for speeds not far from 20 knots. However, on recent armored cruisers whose maximum speed varies from 21 to 23 knots, a slightly greater value of K has been adopted equal to 22.5 kilos."

As in Joëssel's formula, expressed in this form with French units, the value of K is 41.35 kilos; these values correspond to coefficients of reduction of .48 and .54, respectively.

The coefficient .48 seems to be based on the trial of the *Condor* and *Epervier*, which are small vessels of 1,272 tons displacement; the results for larger vessels give for the *Warrior* (8,695 tons, 11.04 knots, helm angle 25 degrees), .62; for the *Rodney* (10,300 tons, 16.9 knots, helm angle 34 degrees 30 minutes), .63; and for the *Amiral Baudin* (12,000 tons, 15 knots, helm angle 35 degrees), .80; which shows that .48 is much too small a coefficient for the ordinary case. M. de Courville, in his article, "Note au sujet du calcul du Moment Resistant des Gouvernails," Bulletin de l'Association Technique Maritime, No. 7, p. 57, gives a mathematical comparison between the results of the *Condor* and the *Amiral Baudin* to show that the coefficients obtained from the latter are what would be expected on account of the greater dimensions of the vessel, but since his method applied to the *Cassini* (960 tons displacement) would give smaller coefficients than for the *Condor*, while the actual results from trial are very considerably larger, the method would seem to be made for the case in point. Comparing the *Rodney* with the *Amiral Baudin* it is to be noted that the former is a longer and deeper ship, has a smaller area of rudder in proportion to the lateral plane, and a smaller coefficient of fineness than the latter, all of which would tend to cause a larger reduction coefficient at the same speed; while an increase of two knots would not of itself account for a reduction in coefficient from .80 to .63. In general, in comparison with other ships, the *Amiral Baudin* seems to have a larger reduction coefficient than would be expected, but this fact shows that for large, slow ships a coefficient of .8 should be used for safety.

Ship.	Length. Feet.	Draft. Feet.	Displacement. Tons.	Area of Rudder. Square Feet.	Speed in Knots.	Helm Angle. Degrees.	Coefficient of Reduction.
Warrior (*a*)	380	26.6	8,695	180	5 92	25	.75
					8 88	25	.73
					11.04	25	.62
Messina (*b*)				31	8.75	40	.50
British Prince (*c*)				101	12.25	35	.68
				104	13.	35	.71
Rodney (*d*)	345	27.25	10.300	195	16.9	34.5	.63
Amiral Baudin (*e*)	328	25.9	12,000	207	15.17	35	.785
					14 72	35	.810
					14.72	33	.802
					14.10	34	.776
Cassini (*f*)	262	10.25	960	53.8	12.	20	.801
					12.	31	.562
					16.	20	.798
					16.	31	.535
					20.	20	.794
					20.	31	.517
Pothuau (*g*)	371	21.3	5,320	.....	17.8	34	.514
					18.2	33	.515
Monterey (*h*)	256	14.5	4,027	95.7	12.	35	.353
					10.	35	.343
					8.	35	.370
					6.	35	.459
					4.	35	.649
Condor and Epervier (*i*)	223	13.9	1,272	.....	6.	9	.792
					6.	18	.723
					6.	28	.704
					8.	9	.730
					8.	18	.650
					8.	28	.630
					10.	9	.675
					10.	18	.584
					10.	28	.570
					12.	9	.620
					12.	18	.528
					12.	28	.520
					14.	9	.565
					14.	18	.480
					14.	28	.478
					16.	9	.515
					16.	18	.443
					16.	28	.443
					18.	9	.465
					18.	18	.414
					18.	28	.412
					20.	9	.420
					20.	18	.389
					20.	28	.385

Breadth of rudder in feet: (*d*) 9, (*h*) 10.41.
Breadth of rudder in feet, assumed from other data given: (*a*) 7, (*b*) 3.4, (*c*) 4.5.
Angular velocity of helm in degrees per second: (*e*) 3, (*f*) about 2 5, (*g*) 2 to 2.5, (*h*) 1.5 to 2. (*i*) 3.
(*a*) Trans. Inst. of Nav. Arch., vol. v., p. 82.
(*b*) Trans. Inst. of Nav. Arch., vol. v., p. 115.
(*c*) Trans. Inst. of Nav. Arch., vol. xxvii., p. 358.
(*d*) Trans. Inst of Nav Arch., vol. xxvii., p. 358.
(*e*) Bulletin de la Soc. Tech. Maritime, vol. vii., p. 57.
(*f*) Bu letin de la Soc. Tech. Maritime, vol. viii., p. 215.
(*g*) Bulletin de la Soc. Tech. Maritime, vol. viii., p. 215.
(*h*) Trans. Soc. Nav. Arch. and Marine Engineers, vol. iii., p. 45.
(*i*) Pollard and Dudebout, Theorie du Navire, Tome iv., p. 36.

No definite reduction coefficients can be stated from the data at present available, but a coefficient must be chosen by judgment for each case, varying from .80 for a large, slow vessel, to .45 for a small, swift one. It would seem best, however, to use from .60 to .75 for modern battleships and cruisers, which is considerably in excess of the figure used in recent French practice (.54) as stated above; .55 to .65 for gunboats of 900 tons and upward; and .45 to .55 for torpedo boats and destroyers.

The pressure on the rudder and the twisting moment about the stock having been estimated, the only member that need be proportioned by calculation, for a rudder as ordinarily hung, is the

stock, as the pintles when proportioned for bearing surface and symmetry will be amply strong to resist any shearing stress to which they may be subjected. In this case the stock should be calculated to resist a torsional stress; in the case of a rudder supported only by the stock the calculations should be for combined twisting and bending, assuming the center of pressure for bending at the center of area; in the case of a balanced rudder supported by the stock and by a pivot at the bottom, the pivot should be calculated for shearing, using half the total pressure as a load, and the stock for twisting and bending, using for a bending moment $\frac{Pl}{8}$, where P = the total pressure on the rudder and l = the distance between the points of support of the rudder. In other words, the rudder is assumed to be a beam fixed at the stuffing box and supported at the pivot.

The formula for the torsional stress in a shaft, $p = \frac{My}{I}$, where p = the allowable stress, M = the twisting moment, y = the distance of the outside fiber from the neutral axis, and I = the polar moment of inertia of a transverse section of the shaft, becomes for a solid circular shaft, solved for the diameter:

$$d = 1.721 \sqrt[3]{\frac{M}{p}} \quad \text{.....(f)}$$

and for a hollow shaft:

$$\frac{d^4 - d_1^4}{d} = 5.093 \frac{M}{p} \quad \text{.....(g)}$$

Where d = outside diameter in inches.
d_1 = inside diameter in inches.
M = twisting moment in inch pounds.
p = allowable stress in pounds per square inch.

For allowable stress on a steel stock, 7,500 pounds per square inch would seem a safe figure, giving a factor of safety of between 6 and 7, which is a customary allowance for structures subjected to live loads. For small, high-speed vessels such as torpedo boats, destroyers, fast yachts, etc., a larger value of p may be used, say 10,000 to 11,000 pounds, owing to the fact that while the stock is calculated for the rudder at a maximum angle at maximum speed, such a condition would never be realized except in smooth water. This fact is also the reason for choosing a safety factor suitable for a structure subjected to a live load rather than for one subjected to shocks. Using 7,500 pounds per square inch for p the formulæ **(f)** and **(g)** become:

$$d = .088 \sqrt[3]{M} \text{ for a solid stock} \quad \text{.....(h)}$$

$$\frac{d^4 - d_1^4}{d} = \frac{M}{1{,}472} \text{ for a hollow stock.} \quad \text{.....(k)}$$

For combined twisting and bending use Grashof's formula:

$$p = \frac{d}{2I}\left(\frac{3}{8} M_1 + \tfrac{5}{8} \sqrt{M_1^2 + M_2^2}\right) \quad \text{.....(l)}$$

Where M_1 = bending moment.
M_2 = twisting moment.
d = diameter of stock.
I = moment of inertia of section about a diameter.
p = allowable stress.

The following example will illustrate the method of making a calculation:

Given a rudder as shown in sketch, maximum helm angle 35 degrees.

Ship data:

L. B. P., 450 feet.
Draft, 24 feet 6 inches.
Beam, extreme, 76 feet 10 inches.
Displacement, 16,000 tons.
Speed, 18 knots.

Rudder must be put hard over from amidships in 10 seconds.

For this case a reduction coefficient of .70 may be used.

Divide the rudder into rectangles as shown, and combine the areas to find the center of pressure of the whole. Applying formula **(e)** for center of pressure, $x = .37\ b$, and taking moments about the axis of the rudder,

AREA		ARM		MOMENT
13.2	×	11.9	=	157.
150.2	×	5.24	=	787.
122.1	×	.98	=	120.
285.5		3.73		1,064

Total area = 285.5 square feet, center of pressure 3.73 feet = 45 inches aft of axis.

Then by **(d)** applying the reduction coefficient

$$P = .7 \times 3.57\ AV^2 = .7 \times 3.57 \times 285.5 \times 324 = 231{,}162 \text{ pounds.}$$

Twisting moment = 231,162 × 45 = 10,402,290 inch pounds.

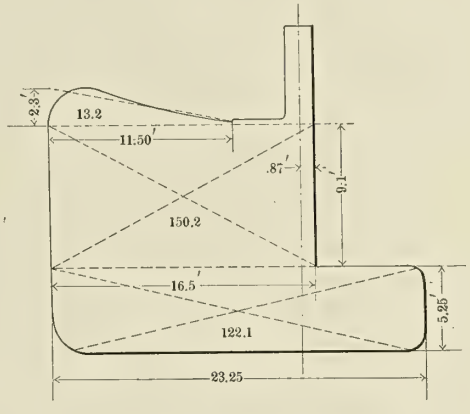
For a solid stock by **(h)**

$$d = .088 \sqrt[3]{M} = .088 \sqrt[3]{10{,}402{,}290} = 19.2 \text{ ins.}$$

The stock is to be hollow with at least a 5-inch bore.

For a hollow stock by **(k)**

$$\frac{d^4 - d_1^4}{d} = \frac{M}{1{,}472} \text{ assume } d = 19.25 \text{ inches.}$$



SKETCH OF BALANCED RUDDER.

Then

$$\frac{137{,}344 - d_1^4}{19.25} = \frac{10{,}402{,}290}{1{,}472}, \quad d_1^4 = 137{,}344 - 136{,}035 = 1{,}309,$$

$$d_1 = 6 \text{ inches.}$$

Therefore, the rudder stock should be 19¼ inches outside diameter, with a bore not exceeding 6 inches.

It will be of interest to compare the above result with the results of various empirical formulæ.

A formula much used in French practice is, for an iron stock, reduced to English units,

$$d = 1.2 \sqrt[3]{KM}.$$

Where d = diameter of stock in inches.
M = moment of area of rudder about the axis.
K = a coefficient which is 1.1 for 14 knots, and varies proportionately with the square of the speed.

This formula applied to the above rudder gives:

$$d = 17\tfrac{3}{4} \text{ ins.}$$

Lloyd's formula is:

$$d = \sqrt[3]{D \times B^2 \times S^2}.$$

Where d = diameter of stock in inches.
D = draft in feet.
B = breadth of rudder in inches.
S = speed in knots.

This formula gives:

$$d = 21 \text{ ins.}$$

A later formula of Lloyd's is:

$$d = \sqrt[3]{D \times b\,(2B - b) \times S^2}.$$

Where b = maximum breadth of rudder in inches.
B = greatest distance from center of pintles to back of rudder in inches.
Other letters as above.

This formula in this case is practically the same as the other. Lloyd's tables, as well as the rules of the American Bureau of Shipping, give 11 inches for diameter of rudder stock for a ship of this number, which shows that these formulæ are not adapted to rudders of this shape, but only to those of merchant vessels where the breadth of the rudder is not nearly so great in proportion to the area.

The rules of the British Corporation for the Survey and Registry of Shipping give:

$$d = .26 \sqrt[3]{R.A.V.^2}.$$

Where d = diameter of stock in inches.
R = distance of center of gravity of rudder area from the axis of the stock, in feet.
A = area of rudder in square feet.
V = speed of vessel in knots.

This would give for the diameter of stock for the assumed case 23½ inches, which shows, as in the two previous cases, the inadaptability of the formula to a rudder of other than the ordinary shape.

Mackrow, page 314, edition 1902, gives a formula for strength of rudder head for a maximum rudder angle of 35 degrees.

$$D = .392 \sqrt[3]{\frac{AU^2d}{K}}.$$

Where D = diameter in inches.
A = immersed area of rudder in square feet.
U = speed of ship in knots, allowing 10 percent slip.
d = distance of center of effort from axis of rudder in feet.

Note—For angles at or about 35 degrees the center of effort is situated ⅛ of the breadth of the rudder before the center of gravity of the area.

K = a coefficient of 5 for cast steel, 4 for wrought iron, and 3 for phosphor bronze.

For this case the above formula becomes:

$$D = .392 \sqrt[3]{\frac{285.5 \times 392 \times 4.08}{5}} = .392 \sqrt[3]{91{,}323}$$
$$= .392 \times 45$$
$$= 17.64 \text{ inches.}$$

Mackrow, page 176, edition 1902, gives a formula for twisting moment on the rudder stock:

$$M = 1.12\, AV^2 d \sin \theta$$

Where M = twisting moment in foot pounds.
A = area of rudder in square feet.
d = distance from axis of rudder to center of pressure as above in feet.
θ = helm angle.
V = 1.1 times speed of ship in feet per second.

This becomes for the rudder, assumed:

M = 835,034 feet pounds, as compared with 862,234 feet pounds from the modified Joëssel formula.

Attwood, Text Book of Theoretical Naval Architecture, pages 265 and 266, uses the foregoing formula and calculates the size of rudder stock with this moment, allowing a stress of 5 tons per square inch for cast steel, which seems rather in excess of the limits of safety.

It is to be observed that this formula for total pressure on the rudder, *i.e.*, $1.12\, AV^2 \sin \theta$, becomes, when V is expressed in knots and is taken as the speed of the vessel, the 10 percent allowance being made in the form of a coefficient, and $\theta = 35$ degrees:

$$1.12 \left(\frac{6{,}080}{3{,}600}\right)^2 A\,(1.1\,V)^2 \times .5736,$$

$= 2.20\, AV^2$, which corresponds to formula **(d)** modified with a reduction coefficient of .62, and is a good formula for the average case.

The formulæ **(b)**, **(c)**, and **(f)** or **(g)**, may be applied with but little more trouble than any of the above, and have the advantage of being more general; if the conditions are such that formulæ **(d)**, **(e)**, and **(h)** may be used the work of applying them is practically no more than in the case of the other empirical formulæ, and a reduction coefficient may be chosen to suit the given conditions.

I might also state a point possibly not perfectly apparent, that the center of pressure of a rudder having a curved outline may be found in the same manner as the center of gravity, by Simpson's, or other rules for integration, by applying the proper factor (as for instance .37 for an angle of 35 degrees) to the squares of ordinates instead of the factor ½.

Finally, while as has been previously stated, this method of calculation may be of use only as a standard of comparison, as far as the size of the rudder stock is concerned, owing to the unknown forces which act on the rudder in a seaway, it is of direct value in proportioning the steering engine, since the power to be furnished by the latter is a function of the maximum twisting moment on the rudder stock, and is not affected by the shock of the waves.

Progress of Naval Vessels.

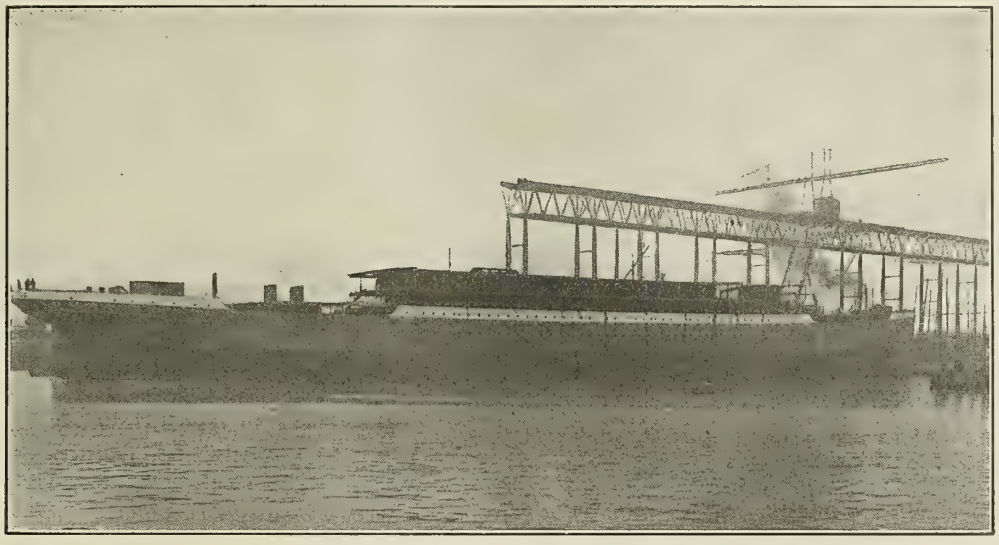
The Bureau of Construction and Repair, Navy Department, report under date of September 11, 1905, the following percentage of completion of vessels building for the United States Navy:

BATTLESHIPS.			Aug. 1	Sept. 1
Virginia	19 knots.	Newport News Co.	91.37	92.89
Nebraska	19 "	Moran Brothers Co.	77.	79 58
Georgia	19 "	Bath Iron Works.	85.	86.44
New Jersey	19 "	Fore River Shipbuilding Co.	87.7	89.3
Rhode Island	19 "	Fore River Shipbuilding Co.	92.1	93.
Connecticut	18 "	Navy Yard, New York, N. Y.	83.67	86.15
Louisiana	18 "	Newport News Co.	82.81	86.4
Vermont	18 "	Fore River Shipbuilding Co.	57.1	61.4
Kansas	18 "	New York Shipbuilding Co.	57.8	60.1
Minnesota	18 "	Newport News Co.	69.9	71.16
Mississippi	17 "	Wm. Cramp and Sons.	34.48	38.71
Idaho	17 "	Wm. Cramp and Sons.	31.22	33.66
New Hampshire	18 "	New York Shipbuilding Co.	15.2	18.2
ARMORED CRUISERS.				
California	22 knots.	Union Iron Works	80.4	81.8
South Dakota	22 "	Union Iron Works.	78.9	80.6
Tennessee.	22 "	William Cramp and Sons.	82.02	84.32
Washington	22 "	New York Shipbuilding Co.	82.03	83.9
North Carolina	22 "	Newport News Co	12.2	14.96
Montana	22 "	Newport News Co	10.81	13.04
SEMI-ARMORED CRUISERS.				
St. Louis	22 knots.	Neafie and Levy Co.	71.4	75.7
Milwaukee	22 "	Union Iron Works.	79.	80.6
Charleston	22 "	Newport News Co.	99.	99.7
SCOUT CRUISERS.				
Chester	24 knots.	Bath Iron Works.	0.	0.
Birmingham	24 "	Fore River Shipbuilding Co.	0.	4.7
Salem	24 "	Fore River Shipbuilding Co.	0.	4.3
TRAINING SHIPS.				
Cumberland	Sails.	Navy Yard, Boston.	95.	95.
Intrepid.	"	Navy Yard, Mare Island	97.5	97.5
TORPEDO BOATS.				
Goldsborough	30 knots.	Wolff and Zwicker.	99.	99.
O'Brien	26 "	Lewis Nixon.	99.	99.
SUBMARINE TORPEDO BOATS.				
Submarine No. 9	.....	Fore River Shipbuilding Co.	12.6	17.6
" " 10	.....	Fore River Shipbuilding Co	11.5	16.5
" " 11	.....	Fore River Shipbuilding Co	11.5	16.5
" " 12	.....	Fore River Shipbuilding Co	11.5	16.5

The New Steamer Aragon.

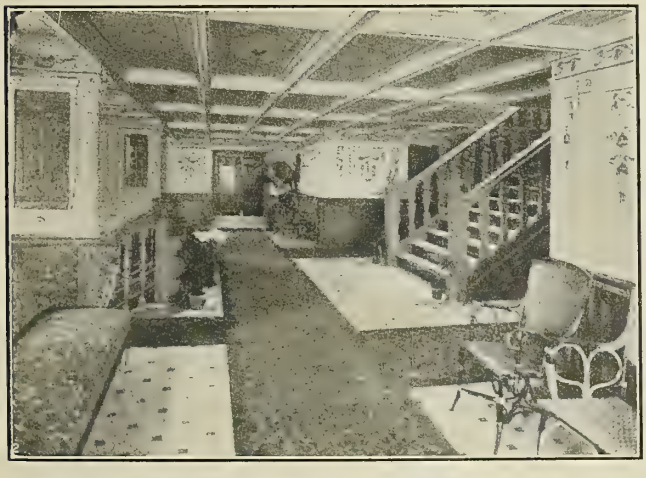
BY BENJAMIN TAYLOR.

We present a photograph of the British Royal Mail Steam Packet Company's new twin-screw mail steamer *Aragon,* recently launched from Harland & Wolff's yard at Belfast.



LAUNCHING OF STEAMSHIP ARAGON.

This steamer is not only the finest vessel in the R. M. S. P. fleet, but will be the largest and finest steamer engaged in the South American trade. Her dimensions are: length, 527 feet 6 inches; beam, 60 feet; gross register, about 10,000 tons. The *Aragon* is designed to carry a large quantity of cargo, but splendid passenger accommodation is provided. The state rooms are on deck, which is an improvement, and many of the cabins are arranged on the tandem principle, with side lights for each room.

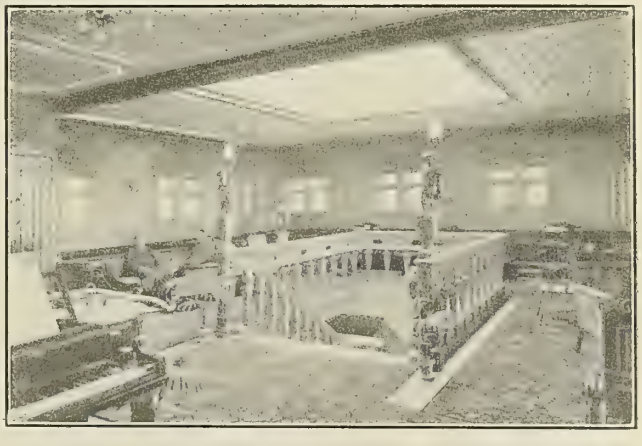


ENTRANCE HALL, WITH STAIRWAY TO PROMENADE DECK.

There are a number of suites de luxe, furnished in white and gold with silk panelings. The first-class dining saloon is one of the handsomest afloat. The tables, seating 210 passengers, are arranged on the latest restaurant system. The elegance of the whole structure, combined with the magnificent stairway and approach to the main deck forms a fine example of marine architecture. The style is renaissance, in white oak and gold furnishings, adorned with cupids and sea trophies. The dome, modeled after that of the Genoese palace, is ornate and brilliantly lighted. The decorations of the library and music room situated on the promenade deck are in satinwood, the structure being surmounted by a dome of stained glass. The smoke room, well ventilated, is on the promenade deck aft, decorated with mural paintings and wainscoted in oak, while two doors open on a veranda extension fitted with tables. A nursery adjoins the first saloon on the main deck. The second-class accommodation, which compares favorably with first-class on the ordinary liner, includes handsome dining saloon, drawing and smoke rooms on deck, and ample promenade accommodation. The third-class accommodation provides for a large number of passengers in a style unsurpassed on any route.

Every provision has been made for the safe navigation of the steamer, and by the adoption of Harland & Wolff's latest balanced quadruple type of engines vibration is reduced to a minimum. The double set of engines for the twin screws constitutes, of course, an additional element of safety. The new ship is a schooner-rigged vessel with two masts, and has graceful lines and a fine appearance. She is a tribute to the high hopes entertained by the R. M. S. P. Company of the future of the South American trade.

By the launch of the *Aragon,* which started on her maiden voyage on June 14, and is almost double the size of the largest steamer hitherto possessed by the company, the directors have given evidence of their determination to keep ahead of the times.



SOCIAL HALL ON STEAMSHIP ARAGON.

One of the practical difficulties they have had to contend with was that they could rely upon only 22 1-2 feet of water at Buenos Ayres, which fact has hampered shipbuilders in designing large vessels suitable for the trade, but the progress of Argentina and the progressive policy of its government justifies the hope that before very long this question of providing a deeper channel for the port of Buenos Ayres will be settled.

THE PASSENGER AND FREIGHT SCREW-STEAMER KONG HAAKON.

BY DR. ALFRED GRADENWITZ.

The screw steamer described in the following has been built by the Eiderwerft Aktien Gesellschaft shipyard of Tönning, Germany, to the order of the Stavanger (Norway) Steamship Company, and is intended mainly for conveying passengers and mail between the various coast stations along the Christiania-Bergen route. Every care has accordingly been taken to serve the comfort and safety of passengers.

The *Kong Haakon*, as the ship is named, has been designed as a hurricane-deck ship, there being a lower deck in both holds. The ice-breaking resistance is provided for in an especially effective manner by intermediate frames running up to the front boiler-room bulkhead, as well as by re-enforcements of the outside planking at the water-line as far as the stern post. Five bulkheads divide the ship into six watertight compartments. Beneath the holds, as well as under the engine, there is a double bottom designed as a ballast tank. The ship has an elliptical stern and a straight stem. The hurricane deck has been made as wide as possible, both in the front and rear parts, with a view to increasing the available room. The ship is rigged as a pole-mast schooner.

On the promenade deck is a teak deck house containing in front the chart room and in the rear a smoking saloon. On the hurricane deck is the iron deck house, to the rear the ladies' saloon, a staircase leading down to the first-class rooms and a larger smoking saloon, while another staircase to the first-class rooms is located in front of the boiler casing. In the two side houses on the hurricane deck are the officers' rooms, two cabins, a post office, and the parcel-post room. Below the hurricane deck is placed to the rear the dining saloon, in front of which a spacious hall as well as a first-class cabin are located, in addition to the captain's and steward's rooms. In the front part of the ship are the second class, and below these the third-class passenger rooms.

All the rooms, including the decks, are lighted by electricity, in addition to which the saloons are provided with artistic silvered oil lamps. All state rooms are provided with steam heating. A projector of 10,000 normal candle power is placed on deck. As to loading and unloading devices, the ship has for the front hatchway a steam winch of 150 mm. (6-inch) cylinder diameter and 250 mm. (10-inch) stroke, the lifting power being 2½ tons, together with a derrick fitted to the mast. For the rear hatch has been provided a steam crane of 2-tons lifting capacity. In addition to this, the ship has a strong windlass, and in the stern a capstan for warping, both being operated by steam. As she is compelled in the fjords very frequently to sail in narrow channels, special care has been

THE SCREW STEAMER KONG HAAKON.

There are 23 first-class passenger cabins on board, containing 78 berths. Moreover, all the saloons are provided with sleeping sofas, so that the ship affords sleeping accommodations for at least 120 first-class, 24 second-class, and 16 third-class passengers. The dimensions of the ship are as follows:

Length between the perpendiculars	59.26 m. (184.0 ft.)
Maximum breadth over frames	8.69 m. (28.5 ft.)
Depth from upper edge of keel to main-deck stringer	4.29 m. (14.1 ft.)
Height from main deck to hurricane deck	2.30 m. (7.5 ft.)
Height from hurricane deck to promenade deck	2.30 m. (7.5 ft.)
Carrying capacity with 4.12 m. (13.5 ft.) draft above upper edge of keel	370 tons
Guaranteed speed with 3.66 m. (12 ft.) draft above upper edge of keel	13.25 knots
Displacement with same draft	993 c. m. (978 tons)
Coefficient of displacement	0.527

Both the stern post and rudder frame are made of steel castings, the rudder being double plated. The bottom of the ship as well as the bilges are lined with cement, all sharp places at the ends of the ship being filled with coke and lined with cement.

bestowed on the construction of the steam steering gear. The ship carries two lifeboats of seamless steel 6.7 m. (22 feet) in length, two wooden lifeboats 6.1 m. (20 feet) in length, and a sailing boat of 5.5 m. (18 feet) length, which are all fitted up according to the regulations of the board of trade.

The main engine has been designed as a triple-expansion engine of the following dimensions:

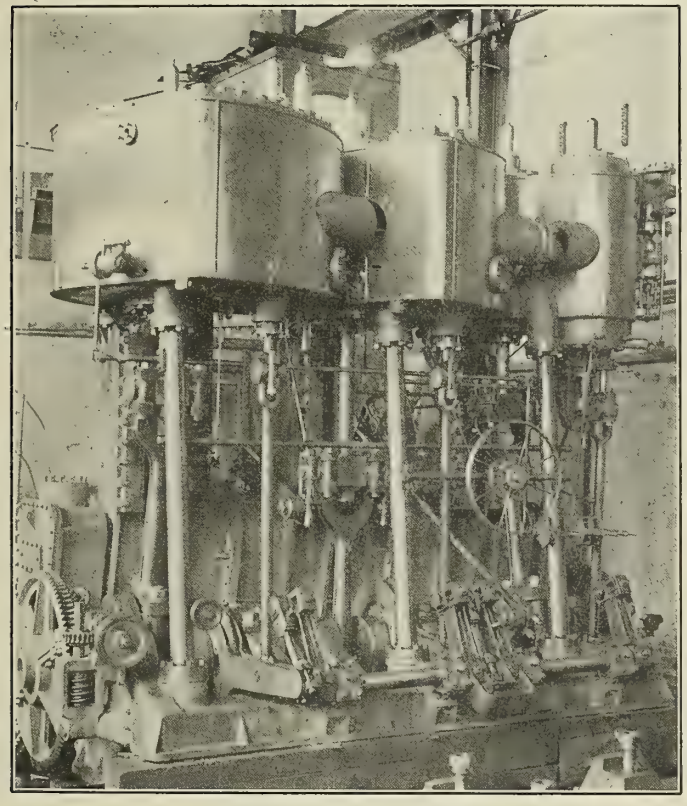
Diameter of high-pressure cylinder	480 mm. (18.9 inches)
Diameter of medium-pressure cylinder	770 mm. (30.3 inches)
Diameter of low-pressure cylinder	1,270 mm. (50.0 inches)
Common stroke of all	750 mm. (29.5 inches)

During the official trials in the Eckernförde Bay a mean speed of 13.52 knots and 1,196 indicated horsepower was recorded with 113 revolutions per minute.

Owing to the shortness of the available engine room the engine has been fitted with Marshall valve motion. The cylinders are substantially screwed to each other and fitted with steam jackets, excepting the low-pressure cylinder. The steam inlet pipe of the medium-pressure valve chest has been designed as a compensation stuffing box, while between the medium- and low-pressure slide valve chests a compensation lens has been arranged. The high-pressure cylinder has been cast in one piece with the corresponding

valve chest, while the remaining cylinders have separate valve chests. The pistons are made of cast iron, and are conical in form.

The surface condenser has a cooling surface of 130 square meters (1,400 square feet), the cooling pipes being of tin-plated rolled brass, 19 millimeters (0.75-inch) in external diameter. On the front part of the condenser, which consists of three parts, is located the reversing engine of 135 millimeters (5.4-inch) cylinder diameter and 130 millimeters (5.2-inch) stroke, operating in the usual way through worm and worm wheel, the weight shaft located on the opposite side. The steel crank shaft consists of three interchangeable parts and is 244 millimeters (9.6 inches) in diameter. The two intermediate shafts, each of which is 5,700 millimeters (18.7 feet) in length are 232 millimeters (9.2 inches) in diameter. The screw shaft is 6,000 millimeters (19.7 feet) in



THE TRIPLE-EXPANSION MAIN ENGINE.

length and 268 millimeters (10.5 inches) in diameter, and runs in renewable cast-iron boxes consisting of two parts and lined with pewter. The screw is made of cast iron to which aluminum is added; it has four blades, is 3,300 millimeters (10.8 feet) in diameter, with a pitch of 4,350 millimeters (14.3 feet); thus, 1.32 is the pitch ratio.

The pumps are fitted to the second third of the condenser and are operated from the cross head of the medium-pressure cylinder by means of a swinging lever. In addition to an air pump of the Edwards system, a double-acting circulating pump, two Lenz pumps, and two feed pumps have been provided, the common stroke of the three latter being 460 millimeters (18.1 inches).

To feed the boiler at sea a Weir pump situated on starboard side is used, which is so designed as to make only eleven double strokes per minute with the maximum output of the engine. The two feed pumps before mentioned serve as reserve. In addition there is on the starboard side a bilge pump of 80-tons hourly output, and a donkey pump, being large enough to feed both boilers under forced draft. The feed pump for the auxiliary boiler, and the injectors, are arranged in the boiler room.

The electric-light engine has likewise been located on starboard side, and consists of a compound engine of 150 and 260 millimeters (6 and 10.3 inches) cylinder diameter, 160 millimeters (6.3-inch) stroke, and 300 revolutions per minute; it is direct-coupled to the direct-current dynamo.

The two main boilers are located one behind the other in the boiler room and are worked at 13 atmospheres (190 pounds per square inch). Each of them has three corrugated furnaces, the dimensions being as follows:

Length	3,445 mm. (11′ 3½″)
Diameter	3,960 mm. (13′)
Diameter of furnaces	980 mm. (3′ 2½″)
Number of tubes	112
Inner diameter of ordinary tubes	80.5 mm. (2.64″)
Outside diameter of ordinary tubes	89 mm. (2.92″)
Number of stay tubes	102
Inside diameter of stay tubes	77 mm. (2.53″)
Outside diameter of stay tubes	89 mm. (2.92″)
Heating surface	148 sq. m. (1,590 sq. ft.)
Grate surface	4.29 sq. m. (46.2 sq. ft.)
Ratio	34.5 to 1

The boilers are of mild Siemens-Martin steel, the strength required for the shell being more than 27 and less than 32 tons per square inch, the extension being at least 20 percent.

THE SALOON, LOOKING AFT.

The auxiliary boiler is on the starboard of the boiler room, and has been designed on the Victoria type for 8.4 atmospheres (123 pounds per square inch), the dimensions being as follows:

Height	2,743 mm. (8′ 4½″)
Diameter	1,600 mm. (4′ 10½″)
Number of fire tubes	77
Heating surface	16 sq. m. (172 sq. ft.)
Grate surface	1.48 sq. m. (15.9 sq. ft.)
Ratio	10.8 to 1

Staybolt Diagram.

BY H. S. BRITT.

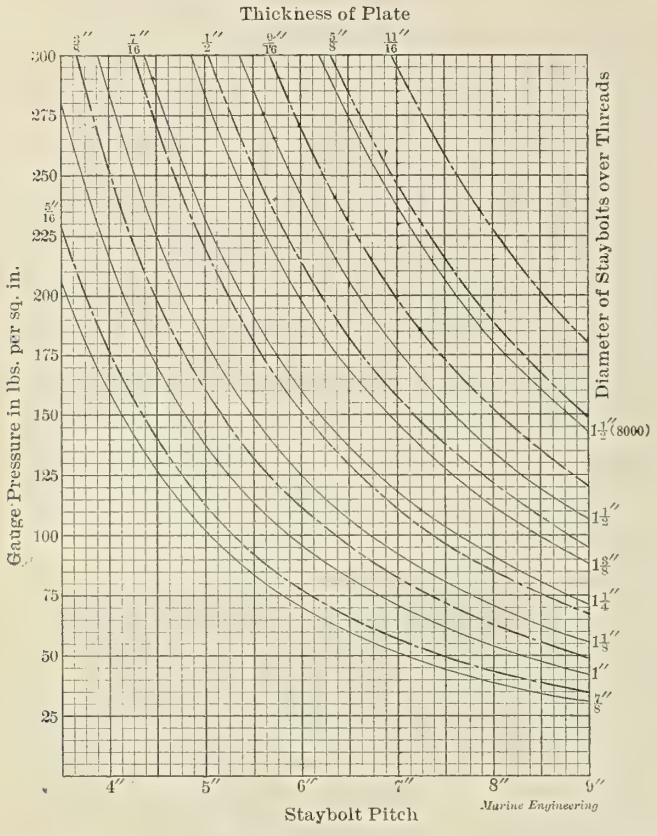
The accompanying diagram embodies the rules of the U. S. Steamboat Inspection Service for flat surfaces supported by screw staybolts. The heavy dot-and-dash curves show the maximum pitch for any particular plate thickness and pressure, as determined by the following formula:

$$P = k \frac{t^2}{p^2},$$

where P = allowable working pressure,
t = plate thickness in sixteenths of an inch,
p = maximum staybolt pitch,

and k is a constant having the value of 112 for plates 7-16 inch thick and under, and the value of 120 for plates above 7-16 inch.

The fine curved lines are for the determination of maximum staybolt pitch as dependent upon the strength of the staybolt.



These lines, with the exception of the one marked 1 1-2 inches (8,000), are plotted for the following conditions: staybolts in rectangular rows pitched the same distance apart vertically and horizontally; stress per square inch of cross-sectional area of a staybolt, 6,000 pounds; 12 threads per inch; sharp V-threads. For the curve marked 1 1-2 inches (8,000) the conditions are the same, except that 8,000 pounds per square inch is used for the unit stress, this higher value being allowed by the United States rules for staybolts over 1¼ inches diameter at bottom of thread if tested by the inspector before being used.

In case the pitch of the staybolts is not the same vertically and horizontally, the greater of the two pitches must not exceed the maximum for the given plate thickness, as found by the use of the heavy dot-and-dash curves; and the product of the vertical and horizontal pitches must not exceed the square of the maximum pitch for the given size of staybolt, as found from the diagram.

Trial Trip of the Sea-Going Suction Dredge St. John.

The dredge *St. John*, which has had, near Sandy Hook, a three-days' successful demonstration of her sea-going and dredging qualities, is the first of two vessels constructed by the James Reilly Repair and Supply Company for the United States Corps of Engineers.

The *St. John* was built to dredge sand from the mouth of the St. John River, Florida. The vessels have wooden hulls and have the general appearance of a coastwise steamship. The suction pipes are 18 inches in diameter, swiveling amidships, and are constructed so that they can be lowered or inclined at the free ends where heavy scraper nozzles are attached. These dredge the bottom of the harbor, and sand is pumped up by means of powerful centrifugal sand pumps which discharge the heavier material into bins constructed in the hold of the vessel. These bins have a capacity of 1,600 tons and can be filled in about forty-five minutes. The dredges are self-propelling, and when under way the suction pipes are drawn up and the vessel proceeds to sea, where the sand is discharged through numerous sluices in the bottom of the hull. The dredges are 200 feet long, 40 feet beam, 23 feet 2 inches depth, and have each two large Scotch boilers furnishing steam for about 2,000 horsepower. The engines are of the vertical inverted compound type, of about 1,200 horsepower. The pumping machinery on each vessel consists of two centrifugal pumps driven by a vertical inverted compound engine of about 700 horsepower. The vessels have in addition the usual auxiliary machinery fitted to regular steamships, and will each carry a crew of thirty-six men. They are lighted by electricity and are generally fitted out in a first-class manner.

LAUNCHING OF THE SUCTION DREDGE ST. JOHN.

MOTOR BOATS, IV.

BY DR. WILLIAM F. DURAND.

THE MOTOR-BOAT ENGINE.—(*Continued.*)

Ignition. For the purpose of igniting the charge, present-day practice is divided between the so-called "make-and-break" or contact spark, and the jump spark. In the former the spark is formed between two metallic points in the circuit of a spark coil, which are brought together and then "broken" or quickly separated. The jump spark, on the other hand, is formed between two stationary points which are in the circuit of the secondary of an induction coil. Contact being then made in the primary coil at the proper instant in the stroke a stream of sparks plays between the two terminals on the secondary circuit located within the cylinder and separated by a distance of about $\frac{1}{16}$ inch. The contact spark requires moving parts within the cylinder, while the jump spark does not. With both forms trouble may arise due to corrosion or wear on the points, or to their becoming covered by deposits of a gummy carbonaceous matter. In special designs of make-and-break devices various means are in use to provide or permit a gradual shifting of the actual sparking surfaces so as to distribute the wear over a considerable surface and thus increase the durability of a given set of "points." With both forms of device the time at which the spark shall pass relatively to the position of the piston may be varied by an adjustment of the connections controlling the operation of the sparking gear. It is usually claimed by the advocates of the "jump" system that this adjustment is more readily effected than by the "make-and-break" system, and that the resultant speed range is wider; and also that the sparking points carried in an easily removable plug are more readily examined or replaced. There seems to be foundation for this claim and it must be admitted for most cases of high speeds and large power, as well as where readiness of handling and range of speed are of importance, that the jump spark is usually employed.

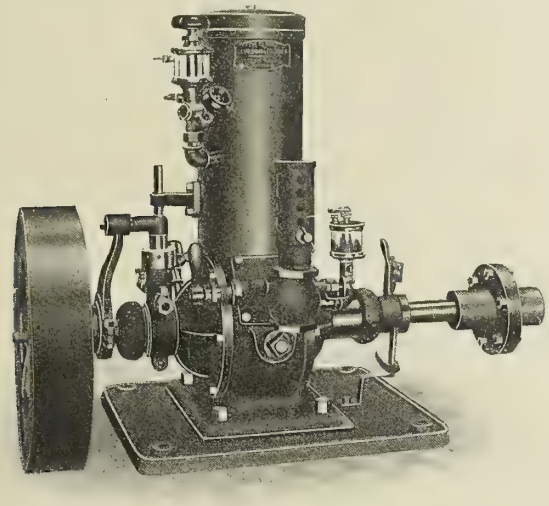


FIG. 28.—SINGLE CYLINDER, TWO-CYCLE "LAMB" ENGINE.

Cooling. Due to the high temperature of the products of combustion, the walls of the cylinder tend to become heated beyond the point of practicable lubrication, resulting in the rapid cutting and scoring of the bearing surfaces of the piston and cylinder. To prevent this the excess heat in the metal is removed by surrounding the working parts of the cylinder by a water jacket, as indicated in the various cuts. In the best practice of the day provision is made for independent expansion between the inner cylinder or liner and the outer cylinder or barrel, in order that the former may expand and contract without constraint from the latter. In many cases of excellent design, furthermore, the jacket is made by a brass shell fitted over the cylinder casting, which is accurately turned on both inside and outside, thus insuring uniformity of thickness in both parts of the structure, uniformity of temperature about the entire circumference of the cylinder, and, in consequence, a true circular form—a result scarcely possible with a cast-iron structure in one casting, with its inevitable variations in thickness from one point to another. The cooling water is usually provided by a small reciprocating plunger operated by an eccentric on the main or valve cam shaft. In some cases, however, rotary pumps have been used with good results. It may be noted at this point that it is possible to carry the cooling action too far as well as not far enough. In the latter case there is, of course, danger of mechanical injury to the piston and cylinder. In the former too much heat may be removed from the cylinder, and its efficiency as a heat engine may thereby be decreased.

Lubrication. For lubrication of the crank pin, designers of the two-cycle engine have found the splash system a simple and effective means. The bottom of the crank case is filled with oil to such a point that the crank will splash the oil as it goes around, and thus receive a supply of lubricant. The same general source is also usually depended on to supply lubricant for the piston and for the upper end of the connecting rod as well. The chief objection to this mode of lubrication is found in the fact that the charge of vapor passing from the crank case to the cylinder will carry with it some small amount of lubricating oil. This in the cylinder will become burned, producing a charred or semi-viscid residue, which in time may clog the passages or cover the sparking points. Due to this fact some of the most recent designs show special means for oiling the cylinders and the parts within the crank case at stated times or at will, and by special adaptation of the means commonly employed in general engineering practice. For the other bearings and moving parts of the engine, compression grease cups, sight feed cups, and other similar means common to the engineering practice of the day are employed.

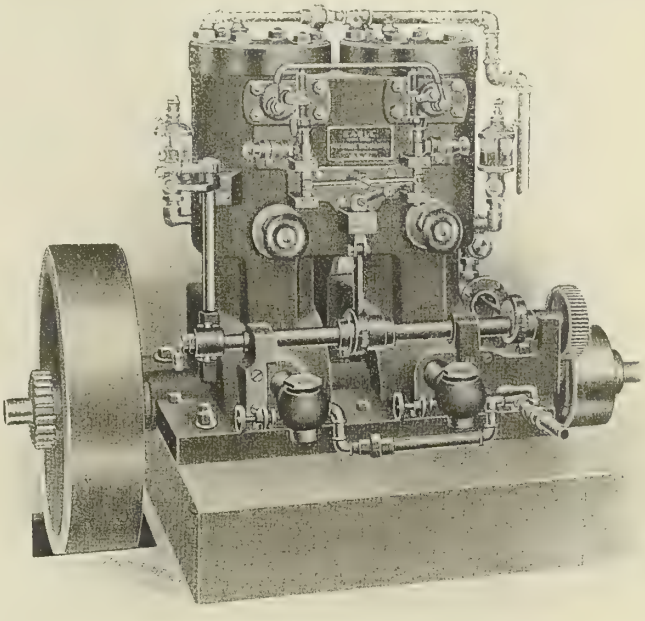


FIG. 29.—TWO CYLINDER, TWO-CYCLE, 10 H.P. "EAGLE" ENGINE.

Speed Control. For purposes of speed control some combination of two general means is employed. These are:

(1) Control by throttling the amount of mixture introduced per stroke, and hence of the resulting work which can be developed.

(2) Control of the instant of sparking relative to the position of the piston, thus modifying the power which will be developed by a given charge. Reference to the form of indicator card given in the introductory chapter and to the description of the operation of the engine will show that at some one point the card

corresponding to a given charge will have the largest possible area. Retarding the spark beyond this point will then give a card of which Fig. 30 may be taken as a typical illustration. In some cases use has been made of an impoverishment of the mixture by increasing the amount of air. This, however, is not to be recommended, as it is less certain and satisfactory than a suitable combination of throttle and spark control.

In cases where the engine is connected to the propeller shaft through a reversing clutch, and where in consequence the engine may be running without load, the throttle, in the best designs, is placed under the control of an automatic governor of the centrifugal type, thus placing the speed under limitation with any and all conditions. Where the engine is multi-cylinder and of the so-called reversing type, the engine is connected direct to the propeller shaft, and is thus under load at all times. In such cases the engine is therefore controlled by hand throttle as in the steam engine.

Reversing and Handling. For reversing the thrust of the propeller with the internal combustion marine motor there are three methods at hand:

(1) Reversing the obliquity of the propeller blades.

(2) Reversing the direction of rotation of the propeller shaft by a mechanical reverse located between the engine and the shaft.

(3) Reversing the direction of the engine itself.

Further reference to Number (1) will be deferred to the general topic of the propeller.

The mechanical reverse gear is usually some form of the well-known "box of gears" or "jack-in-the-box" type. It operates in such manner that when in the go-ahead motion the engine is coupled direct to the propeller shaft, and the entire arrangement operates as a solid coupling, and, of course, without waste of power. When in the reverse motion, the engine drives the shaft through the gears, by the operation of which it is reversed in direction, and with the resulting slight loss due to the friction and wear in the gears. For engines of small or moderate size this form of reverse has generally commended itself to designers, and in one form or another is quite commonly employed. In large sizes, however, it is preferable to reverse the engine itself, as in the steam engine, and in certain cases of recent design the reverse is effected in this manner. It is clear that an internal

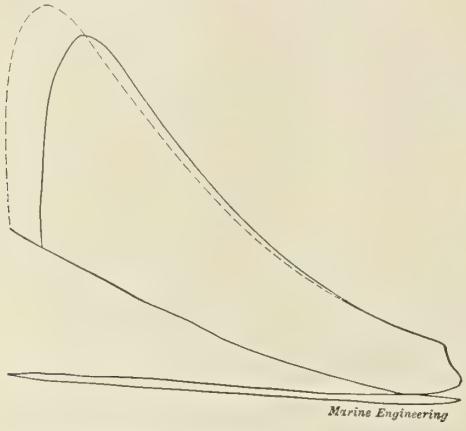


FIG. 30.—INDICATOR CARD WITH RETARDED SPARK. DOTTED LINE SHOWS NORMAL CARD.

combustion engine is as ready to run in one direction as the other, and that it requires only the proper adjustment of valve gear and sparking instant, together with an initial impulse, to start in either direction. In such form of reverse, then, the angular position of the cams operating the valves and the spark advance are both under lever control in such manner that from a mid-position the lever may be moved in either direction, thus placing the valves and spark in suitable relations for operating in one direction or the other according to the position of the lever. The initial impulse may then be obtained in the usual manner for starting and the engine will continue in the same direction. While this mode of reverse requires some additional features in the mechanism of the valve gear, still it must be considered as the most satisfactory in general principle, and in operative detail, except perhaps for engines of the smallest size. It may be added, however, that in order that this mode of reverse may be able to demonstrate its full value for rapid backing and filling, it should be used in connection with an auxiliary starting arrangement of some effective type. A typical arrangement of this character includes a small auxiliary air-compressor pump and receiver, with throttle and connections, with the compression space in the cylinders of the engine. At the will of the operator the compressed air may then be turned into the cylinder or cylinders just near or beyond the point of maximum compression, and thus the engine may be started as readily and in the same manner as with steam. The compressor may be so arranged as to throw in or out of gear when desired, and the receiver may readily be made of sufficient capacity to serve for starting the engine from 20 to 30 times. In the handling of engines of smaller size where the auxiliary compressed-air receiver is not fitted, recent practice shows various improvements in detail, such as the provision of a lever for reducing the compression when turning by hand, retaining just sufficient to insure ignition. Manipulation of this lever to reduce the compression serves also to retard the spark, so that there can be no "kick back" or danger to the operator when "cranking" by hand.

Muffling. Great improvements have been made in recent years in connection with the decrease of the sound occasioned by the exhaust. In principle the muffler operates by splitting up the individual puffs, in part mechanically and in part by the acoustic influence of an intermediate chamber, in such manner as to dissipate and absorb the sound energy of the exhaust and reduce the succession of sharp exhaust puffs into a more nearly constant and gentle discharge. It is not practicable to realize the full beneficial effect of the muffler except by some increase in the back pressure against which the engine exhausts, and in cases where the maximum power is an absolute necessity the muffler should be cut out and the noise accepted as the price to be paid for the additional power. To permit of this the muffler should be so installed that it can be readily cut in or out, and in both cases the exhaust leads should be direct and free in order to reduce the back pressure to a minimum.

Economy. The matter of economy is little thought of in connection with the typical engines used in small runabout boats, but with those of larger size—from 20 horsepower upward for example—the question of economy will affect in a serious manner both the expense of operation and the radius of action. It is for these reasons among others, as already noted, that the four-cycle engine, with its superior economy, is preferred for such cases. The formula very commonly quoted with reference to the economy of internal combustion motors using gasoline is "one pint per horsepower hour." This is better than two-cycle engines can be depended on to give, and small four-cycle engines will not give any such figures except with care and under the best conditions. One pint per horsepower hour means good conditions with the four-cycle engine of moderate to large size. At the same time, with special care, this result may be slightly bettered, while on the other hand but slight departure from the conditions for best operation will serve to carry the consumption above this figure. Economy with an internal combustion engine depends on the following chief conditions:

(1) The proper proportion of air to gasoline vapor: *i.e.*, the proper quality of the charge.

(2) A proper degree of compression.

(3) Prompt ignition and complete combustion.

(4) The highest practicable degree of expansion on the power stroke.

(5) Only such an amount of cooling by the water jacket as may serve to keep the surfaces of cylinder and piston in good working condition.

(6) Valve ports of sufficient area to insure prompt and full admission and release.

With these various conditions fulfilled in good degree, and with engines of fair to large size, a consumption of one pint per horsepower per hour may be counted on with confidence, but with neglect of the proper precautions, and in small sizes especially, the consumption may rise to twice this amount, or more.

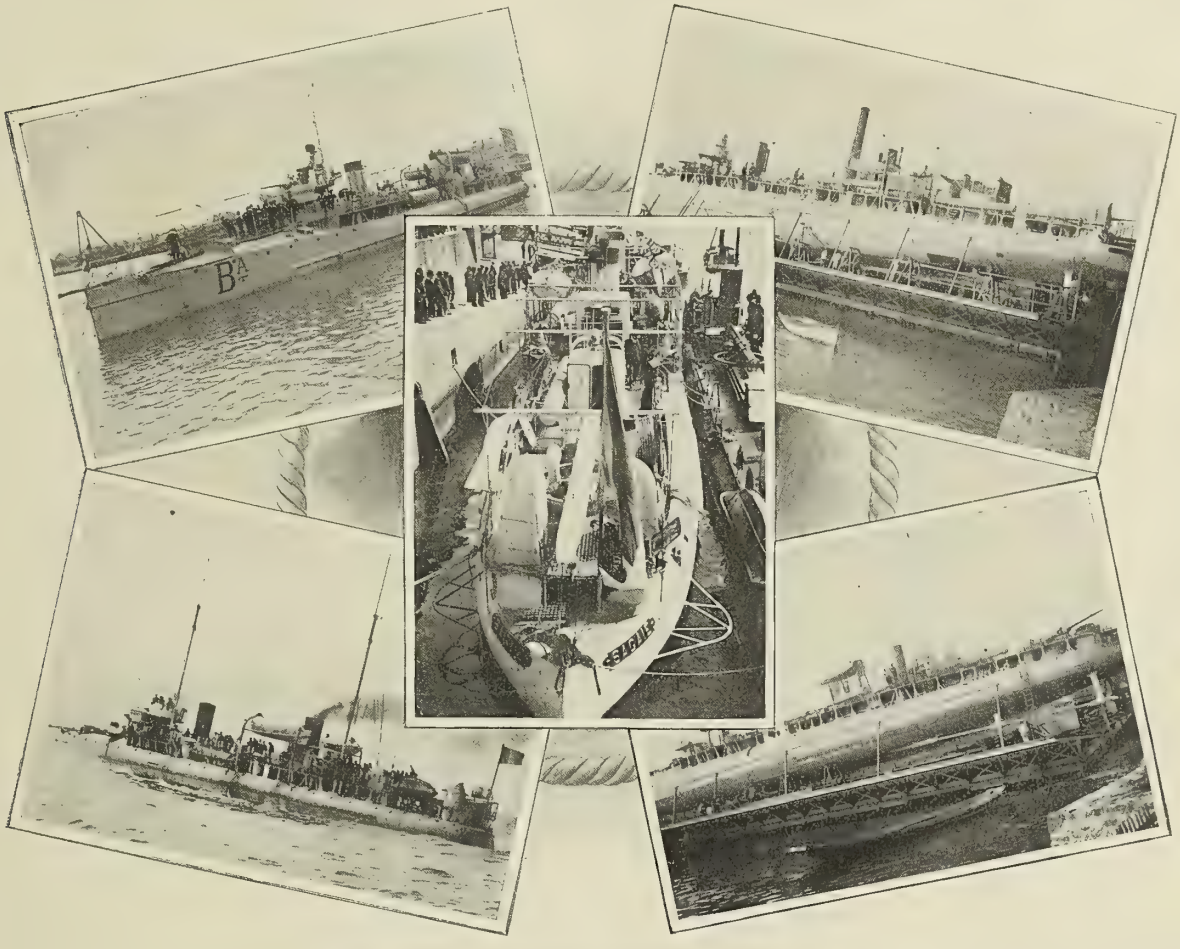
(To be continued.)

French Destroyers.

The destroyers ordered under the last building programme of the French navy are now nearly all in service, and a comparison can readily be made as between the several types. The three types of boat constructed have the same general appearance, being all from the designs of Normand, of Havre. The hulls are so formed that instead of cutting through the water by main force they are

NEW TYPES OF FRENCH DESTROYERS.

Class type	*Durandal.*	*Pique.*	*Mousquel.*
Number	Four.	Nine.	Twenty-three.
Displacement	301 tons.	319 tons.	303 tons.
Length, L. W. L.	180′ 6″	185′ 8″	184′ 8″
Beam, extreme	19′ 6″	19′ 7″	19′ 8″
Draught	9′ 10″	9′ 11″	9′ 4″
Boilers	4 Normand.	4 Normand.	2 Normand.
Engines	2 triple exp.	2 triple exp.	2 triple exp.
Horsepower	4,800	5,700	6,300 to 7,200
Designed speed	26 knots	26 knots.	28 knots.
Trial speed	27 to 27.5	25.5 to 26	30.79
Normal coal supply	38 tons.	34 tons.	27 tons.
Steaming radius:			
At full speed	217 miles.	194 miles.	217 miles.
At 10 knots	2,300 "	2,055 "	2,300 "
Coal consumption at 14 knots speed	.....	.....	910 pounds per hour.
Corresponding radius	.....	.....	900 miles.



NEW FRENCH DESTROYERS OF SECOND AND THIRD TYPES.

expected to glide over the water in much the same manner as some of the extreme types of recent motor boats.

The destroyers form a very small target, as the freeboard is low and the open-work deck above the main part of the hull will explode shells without their striking the main deck. The Normand boilers are quite remarkable in the fact that although the entire power is 7,200 horsepower there are only two boilers in each boat, which means that each boiler must develop 3,600 horsepower. The maximum speed corresponding with the power mentioned is 31 knots, but it is found that the hull vibrates considerably at speeds above 23 knots.

Among the defects of this type may be mentioned the fact that the extremely shallow draft due to the form of hull above mentioned precludes the possibility of attaining high speed in all weather and it is also found that the stability is not so great as might be desired, on account of the high position of the nine-pounder gun mounted on the conning tower. The remainder of the battery consists of six one-pounder guns and two 18-inch torpedo tubes. The crew consists of four officers and 40 or 50 men.

J. Peltier.

Professor P. F. Walker, who has been at the head of the Department of Mechanical Engineering in the University of Maine, Orono, Me., has been called by the University of Kansas, Lawrence, Kan., to take charge of the Department of Mechanical Engineering.

Charles Green, who has had some twenty years of experience covering nearly every important shipyard in the United States, has just opened a consulting office at 2115 Durand avenue, Berkeley, Cal.

Welding Break in Stern Frame of Steamship Apache.

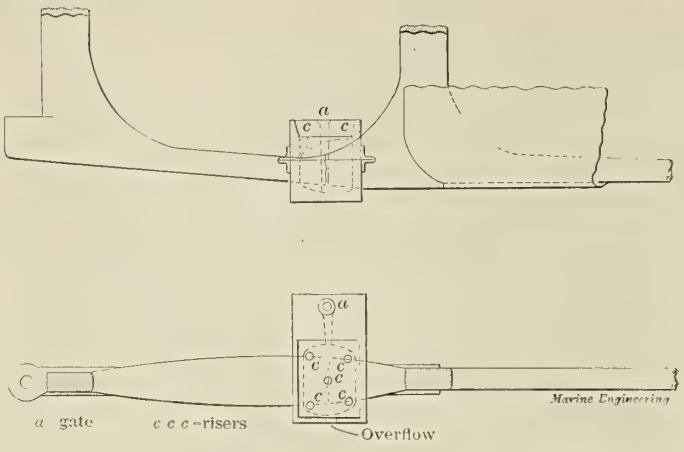
The vessel was placed in dry dock about 1 P. M., July 28, 1905, and the water was entirely pumped out at 2.15 P. M. Propeller wheel was removed and shaft drawn into the hull, for the purpose of examining same for the benefit of underwriters. It was not necessary to remove the wheel and shaft to perform the work. The shoe, or skeg, as it will be hereafter called, had been broken and patched; two large plates had been placed, one on each side, about 2½ inches thick each, and about 16 inches wide. Top and bottom plates had also been put on, forming a box, as it were, around the break. These were removed at 3.20 P. M. The break was found to be about 18½ inches from the after side of what is termed the "stern post" (not the rudder post).

A mold and mold box had been previously prepared, allowances were made in the clay mold so that it could be fitted tight to the skeg. It was decided to have an aperture of clearance in the break for the metal to flow through, and as it was impossible to spread the broken parts, a series of holes, 1 inch in diameter, was drilled vertically through the skeg, nine in number, following the line of the crack. The section to be welded measured 6¾ inches vertically, by 11 inches horizontally. Air drills were started at these holes as soon as the patches were removed. Arrangements were made to have the mold and mold box put into a

PREPARING THE MOLD FOR THE THERMIT WELD.

furnace at about 6.30 P. M., to be left there until the following morning. It will be understood that the mold had been one week in preparation and drying, and the final heating was now to be applied. The skeg was then heated to a red heat and the mold box taken from the furnace and put in place. A foundation was built under the mold box, and a riser of brick, forming a base with a wall about one brick wide and 12 inches high, was formed around the mold box, forming a basin, as it were, into which moist sand was poured. The joints were well looted and sand well packed up around the looting, so that the entire mold was thoroughly protected in case of any leaks. As the section to be welded was very broad it was thought best to have it well ventilated, and five risers were prepared, the riser box being very shallow, with a spout for the slag to run off, and a trough in the sand leading to a pit which was formed at one side, over which was a cover of sheet iron to protect the operators from the heat.

A small open furnace had been prepared and placed on top of the mold, in which a charcoal fire was burnt. Air from the compressed air supply on the dock was utilized for draught, and hot gas was forced down through the gate and up through the riser. While this was in progress, the crucible was put in place, the ordinary tripod being used; and 475 pounds of thermit placed in it, together with 100 pounds of steel punchings previously heated, and 10 pounds of manganese mixed into the thermit. The



SKETCH SHOWING ARRANGEMENT USED.

mixture of the thermit, punchings and manganese was performed by pouring in each consecutively, each being well stirred as it was placed in the crucible.

Owing to the pre-heating, the final reaction was retarded until 4.35 P. M., when the thermit was ignited and the reaction took place. The slag, though precautions had been taken to contain it within walls and canals, overflowed its bounds in the gate, causing some little difficulty for a few minutes; this portion of the slag was covered with sand. Thermit was then used in feeding to the risers and gate, feeding in gradually, maintaining continuous reaction for several minutes to relieve the mold of gases which might form in the body of the weld. This proved to be very effective, as much gas was seen to come from the mold through the risers and gate during the feeding process. This was continued until 50 pounds had been consumed.

THE WELD COMPLETED, BUT NOT TRIMMED.

The mold was allowed to remain in place over night, gradually cooling, and was then annealed by surrounding the weld with a box, into which charcoal fire was prepared. This fire was continued about five or six hours, allowing the weld to gradually cool. In the morning when the mold boxes were removed, the gate and risers were trimmed off, and the grain of the metal appeared to be of exceptionally good quality, and cut very freely with the pneumatic tools. The shaft and propeller wheel were then put in place, and the vessel was ready for docking by 10 A. M. of the third day.

In this work a No. 10 crucible was used, with the ordinary sheet iron cover with an asbestos lining. After the reaction in the crucible the cover, etc., were in good condition, and may be used

for several reactions more. The sheet iron stand which formed part of the crucible was partially melted off, but this does not affect the strength of the crucible, as another stand can be easily applied.

THE NEW AUTO-BOAT NINA UNDER WAY.

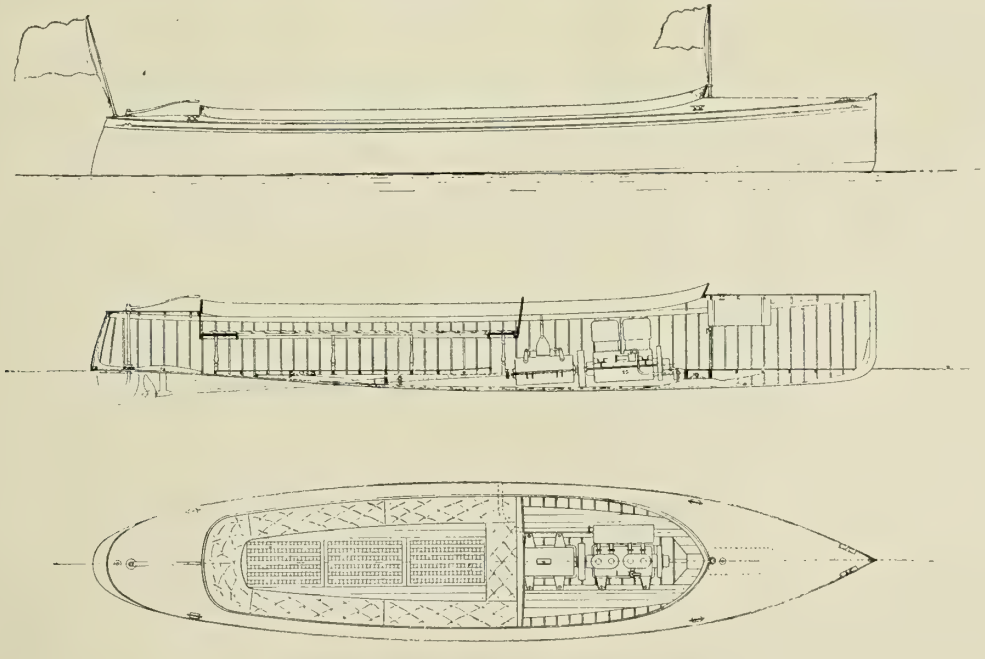
An Interesting Pleasure and Racing Launch.

BY L. RAMAKERS.

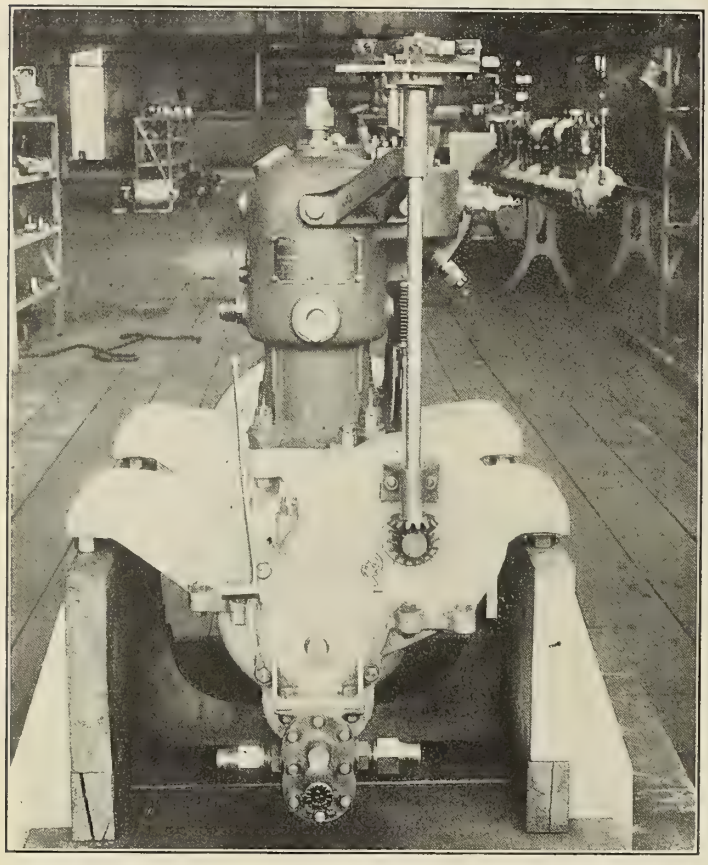
The illustrations we give herewith refer to the English 25-foot fast pleasure or racing launch *Nina*, which, although of very recent make, won last season, during which she took place in several races, the Cowes Cup, the Godfrey Baring Cup (Shanklin), Shanklin Town Cup, and a silver fruit dish at Southsea.

General description.—Beam, 5 feet 1½ inch; draft, 17½ inches. The seams being secured by Saunders sewn or copper stitched system, no rivets or nails are used. The launch is fitted with a four-cylinder 4 by 4½-inch Thornycroft petrol motor developing 20 B.H.P. at 900 revolutions. The engine is provided with high tension magnet and high tension coil and battery ignition. Reversing is effected by means of a lubricated multiple disc friction clutch and epicyclic gear. A large reserve petrol tank is provided under the after seat and supplies petrol to a small tank situated under the turtleback forward of the engine by means of a Wilcox semi-rotary hand pump. Feed from the small tank to the carbureter is by gravity. A speed of 13 miles can be maintained for a ten-hours' run; when in racing trim she can reach about 17 miles. The seating capacity is sixteen persons, if necessary. For racing purposes, the engine and the helmsman are protected by a collapsible canvas hood having a talc window fitted in the front. The whole of the control mechanism can be easily operated by one man, without altering his position.

Motor.—The engine is a vertical water-cooled enclosed motor, designed to run at a normal speed of 900 revolutions per minute, and capable of acceleration to 1,400 revolutions per minute. The cylinders are cast in pairs, each pair forming with its water jacket a single casting. The water jacketing extends about halfway down from the firing end and has been carefully designed to avoid "pockets," and to allow a free and natural run for the cooling water, which is circulated around the cylinders and radiator by a gear-driven positive-action enclosed pump operated from the crank shaft. Each piston is fitted with four cast-iron spring rings, thus obviating any chance of gas leaking past the piston. The



PROFILES AND DECK PLAN OF AUTO-BOAT NINA.



END VIEW OF THE ENGINE.

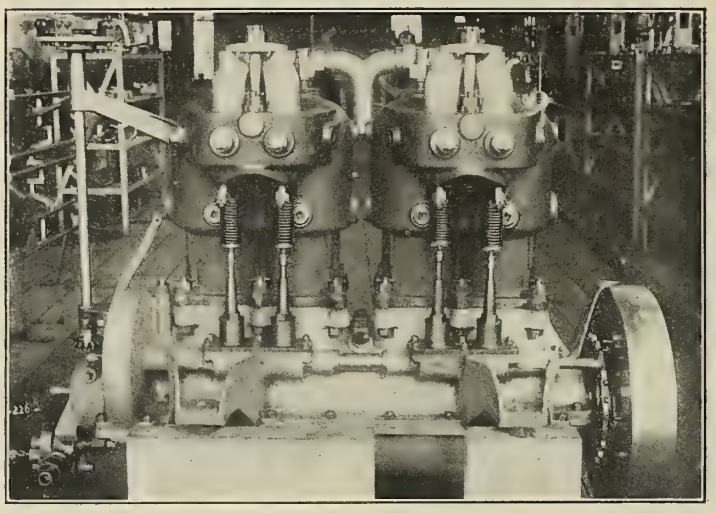
connecting rods are of double tee section, stamped in steel, with gudgeon and big-end bearings, phosphor-bronze bushed, the big ends being adjustable, the big-end cap brasses are slightly less in width than the bottom brasses, thus saving a little weight and ensuring better crank-pin lubrication.

The crank shaft is of special steel, in one piece, with a central bearing between pairs of cylinders; the cranks of each pair of cylinders are at 180 degrees; those of the one pair being at 180 degrees with those of the other pair, thus giving the engine a perfect balance; the end bearings are each 2¾ inches long by 1⅜ inches diameter, thus ensuring ample bearing surface. The forward end of the crank shaft carries the gear wheel operating the water circulating pump above mentioned.

The starting is effected by means of a ratchet wheel and pawl bolted to the flywheel, and a starting handle, secured by a bracket, in an accessible position immediately above the flywheel; the ratchet wheel and starting handle are connected by an endless chain. The crank chamber is of aluminum and is completely enclosed, the internal lubrication being by the "splash" method.

The half-speed shaft is actuated by the crank shaft, and bears the cams for operating the exhaust valves. Between the cams and the ends of the valve stems hard steel rollers are inserted to diminish friction and wear. The exhaust-valve stem guides are very long and of bronze, giving steady support and preventing loss of oil.

The centrifugal governor is carried on the crank shaft and is wholly enclosed in an aluminum casting; it is connected to the throttle valve in the induction pipe and regulates the speed of the motor by throttling the mixture. The water circulating pump is driven by reduction gear from the crank shaft. All gearing is enclosed and lubricated from the crank case. The induction valves are of the spring operated type, and are placed above the exhaust valves on the same side of cylinders, thus tending to keep the latter valves cool. Both induction and exhaust valves are so arranged as to be readily accessible for examination, cleaning, and re-grinding, when necessary.



SIDE VIEW OF THE ENGINE.

UNITED STATES DREDGE BARNARD.

BY ROBERT S. RILEY.

The United States dredge *Barnard* externally is what the Mississippi River men call a good-looking steamboat. Internally she has many more triple-expansion, surface-condensing engines, and other complications, than any river boat ever possessed before. She was designed to help open and maintain the proposed new channel for navigation through the delta of the Mississippi River. South Pass is proving too narrow and tortuous for modern ships, so the government has undertaken to open up the wider channel which Southwest Pass will afford as soon as the bar at the mouth can be cut through.

The *Barnard* was built by the New York Shipbuilding Company, of Camden, N. J., in 1904. Her appearance, finish, and satisfactory performance reflect just credit to her builders.

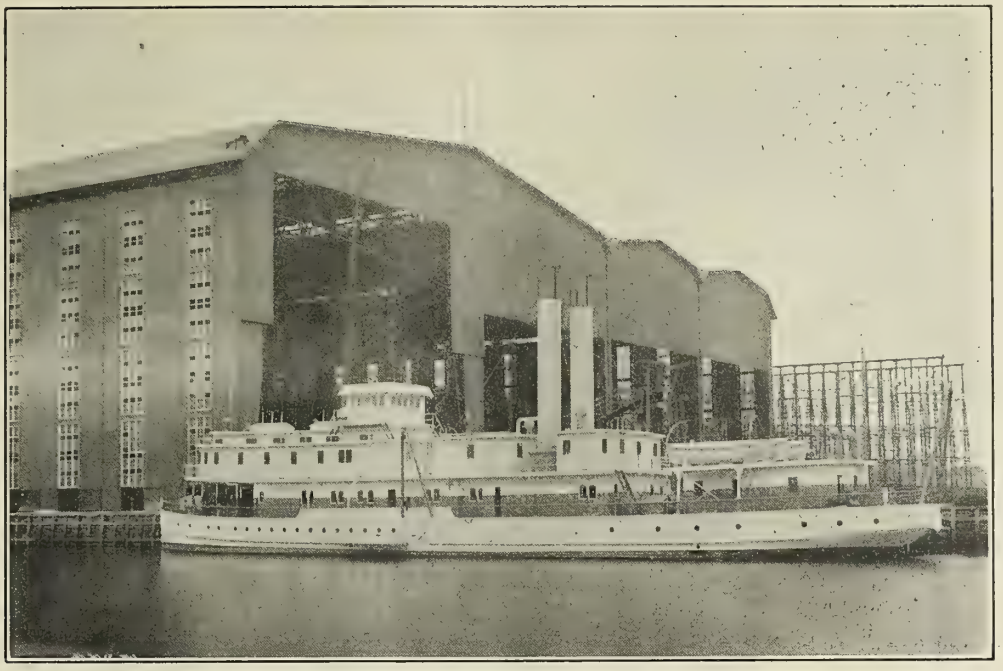
Her principal dimensions are:

Length between perpendiculars	198 feet 6 inches
Length over all	210 feet
Beam	38 feet
Depth molded	14 feet 6 inches
Draft	8 feet

These dimensions, together with the deck plans and illustrations, show that the *Barnard* is a hydraulic dredge of rather an unusual type. She is, in fact, somewhat of an experiment in regard to the arrangement of her dredging machinery, and its combination with the propelling machinery. She combines many features of the Mississippi River hydraulic dredge with those of a light-draft, twin-screw river steamboat. But unlike the up-river dredges, her suction pipe and handling gear are self-contained. This gives her sea-going qualities not possessed by them, and with her speed of 10 knots, makes her quite a handy ship for working over a bar in a moderate sea. She shows less dredging gear on the outside than the sea-going hopper dredges with suction pipes hanging outside. The *Barnard,* however, is not intended for outside use; her work will be mainly in Southwest Pass and the bar at the mouth of the jetties. But she has amply proven her sea-going stability by a trip under her own steam to New Orleans from the New York Shipbuilding Company's yard on the Delaware.

The hull, being shallow, has only the main deck. Heavy iron stanchions support the superstructure, which is entirely of wood. The main feature of the hull is the suction well. It begins 28 feet from the bow, is 7 feet wide, and runs for 60 feet along the midship line. The details of framing, etc., are shown in the cross section of the ship. There are in all seven transverse bulkheads.

The fore peak is utilized as a paint locker. Next comes the chain locker, extending across the ship. From the after bulkhead



THE DREDGE BARNARD AT YARD OF NEW YORK SHIPBUILDING COMPANY.

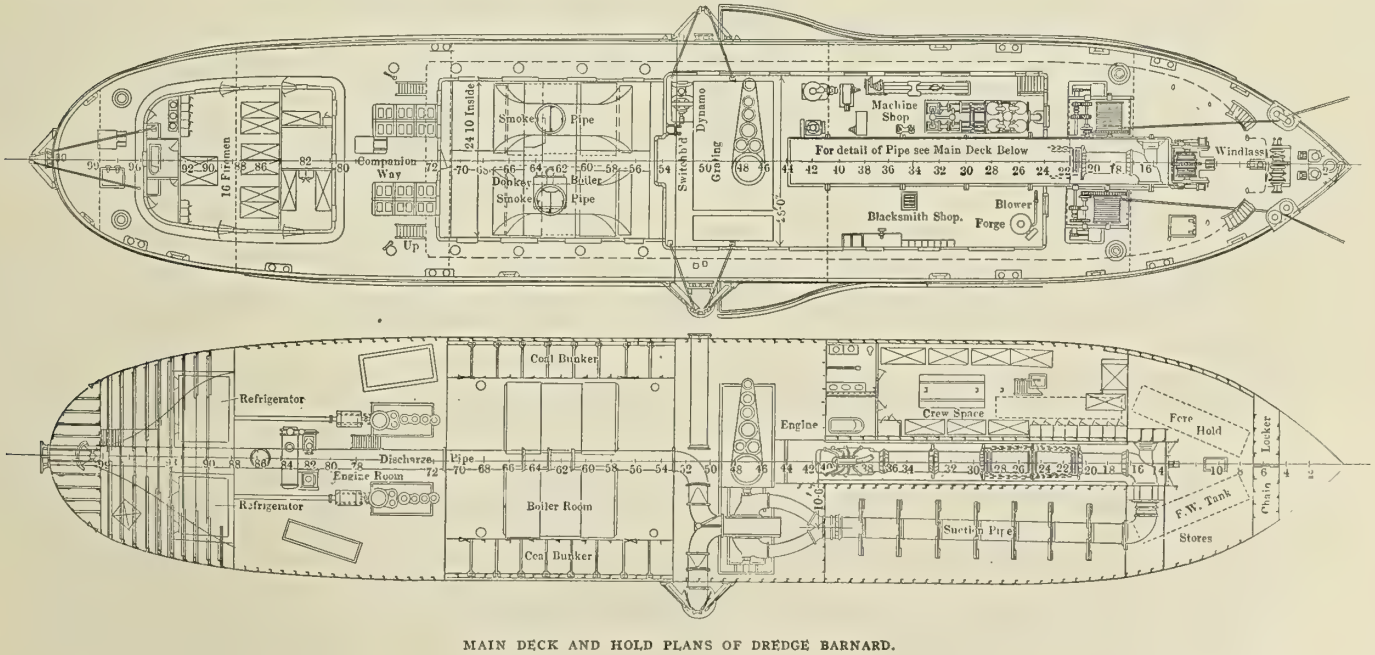
of this to the beginning of the suction well is a store room, beneath the floor of which are stowed two cylindrical fresh-water tanks 40 inches in diameter and 20 feet long.

Abreast of the suction well there are two compartments. The one on the port side is fitted up for the quarters of the deckhands. It has cots for sixteen men. There are also large individual lockers with wash room, bath room, etc. In the center is a table and seats which may all be raised up to the deck above, leaving a clear space for the men. Below the floor of the living quarters is another fresh-water tank of the same dimensions as those under the store room forward of the well. The compartment on the starboard side of the suction well is used for storage of engineers' supplies. The main suction pipe passes through here on its way to the pump.

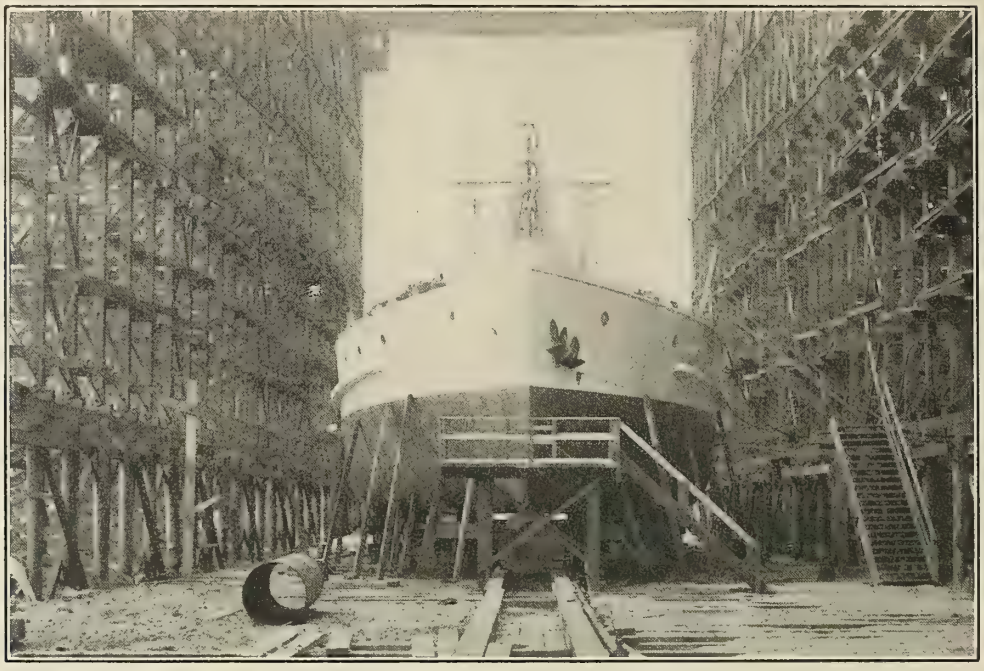
Aft of the suction well is the pumping engine room, extending clear across the ship for 18 feet fore-and-aft. Then comes the boiler room with two double-ended boilers side by side, and bunkers abreast of these. Aft of this comes the propelling engine room for 34 feet, full width. The after end of the ship is occupied by a tank of 22.3 tons capacity, and above it are the cold-storage rooms and stewards' supplies.

The main deck forward is clear of any house for 48 feet, though the superstructure deck is carried out over it to within 14 feet of the bow, where two curved stairways lead up from the main deck. The anchor windlass is located forward. The suction hoisting engine is just forward of the well opening; and the large hauling winches are abreast of the well on either side, just forward of the house. Two capstan barrels also appear on this deck outboard of the hauling winches. Their engines are located beneath the deck, and are accessible from the compartments alongside of the suction well.

The house on the main deck begins 48 feet from the bow, and extends for 96 feet aft, leaving a gangway 5 feet wide along the rail on either side. The forward end of this house contains the

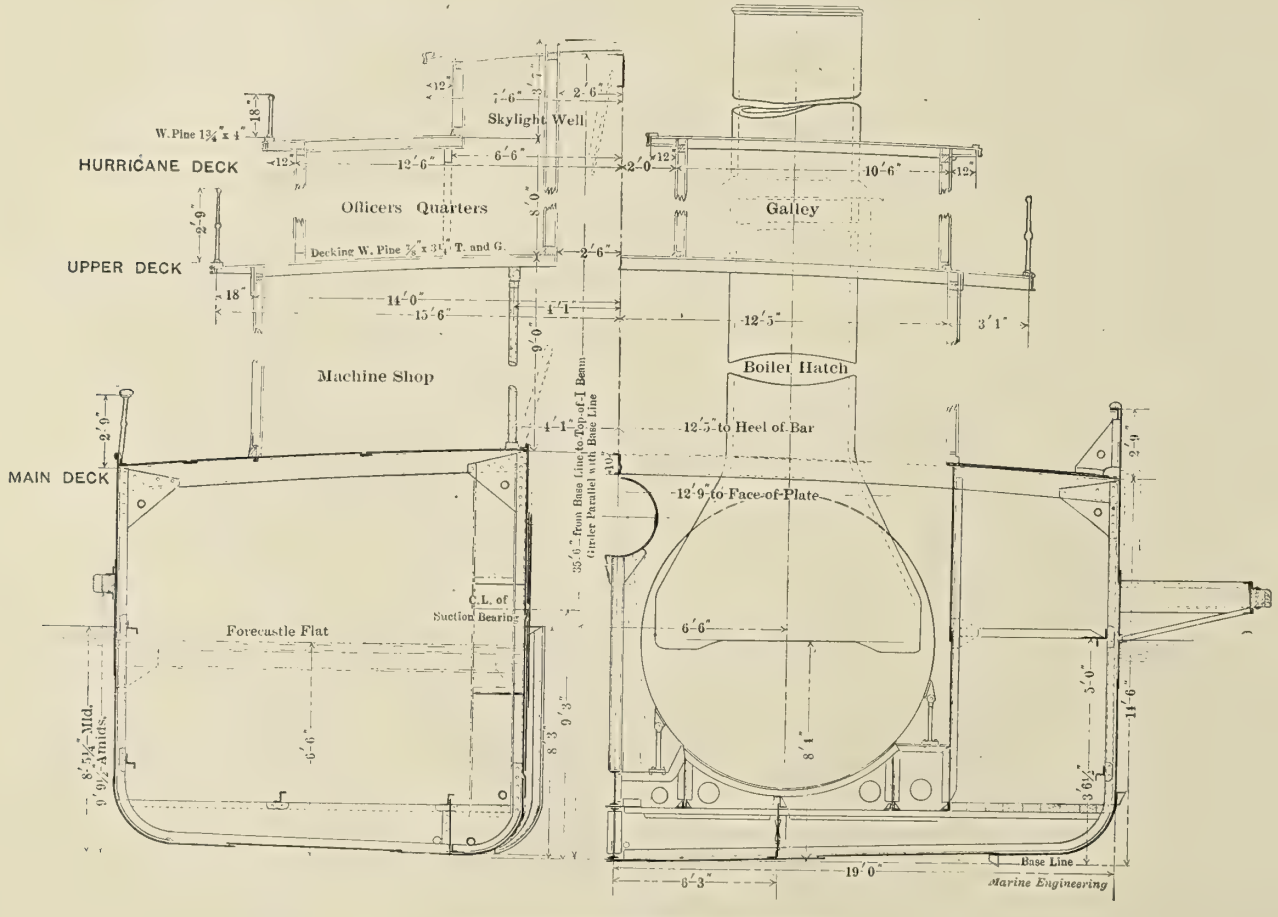


MAIN DECK AND HOLD PLANS OF DREDGE BARNARD.



BOW VIEW OF BARNARD JUST BEFORE LAUNCHING.

machine shop and jet pump on the port side of the well, and the blacksmith and carpenter shops on the starboard side. Then comes the working platform of the pumping-engine room, with the generating set and steering engine. A bulkhead then divides this part of the house from the space over the main boilers. The auxiliary boiler is situated over the starboard main boiler, and a small auxiliary bunker over the port boiler. The house on this deck stops at the after end of the boiler room. There are three large double sliding doors along each side of this house, two at each side of the machine shop, and one each side of the boiler space. These doors, together with numerous large windows, make this deck as comfortable as possible in a tropical climate.



MIDSHIP SECTION OF SUCTION DREDGE BARNARD.

The overhang of the superstructure deck also serves as efficient shelter from sun and rain. Aft of this main house the deck is cut by a companionway and skylights over each propelling engine. Here also two stairways lead up to the after end of the superstructure deck.

The after house on the main deck has a well-equipped laundry on its forward side. The main room is fitted up for the accommodation of sixteen firemen. They have comfortable iron cots, individual lockers, and their own wash room, bath rooms, etc., in the after part of the house. The space in the main deck aft of this is used for handling lines. There are two capstan barrels with engines below the deck; a companionway leads to space below.

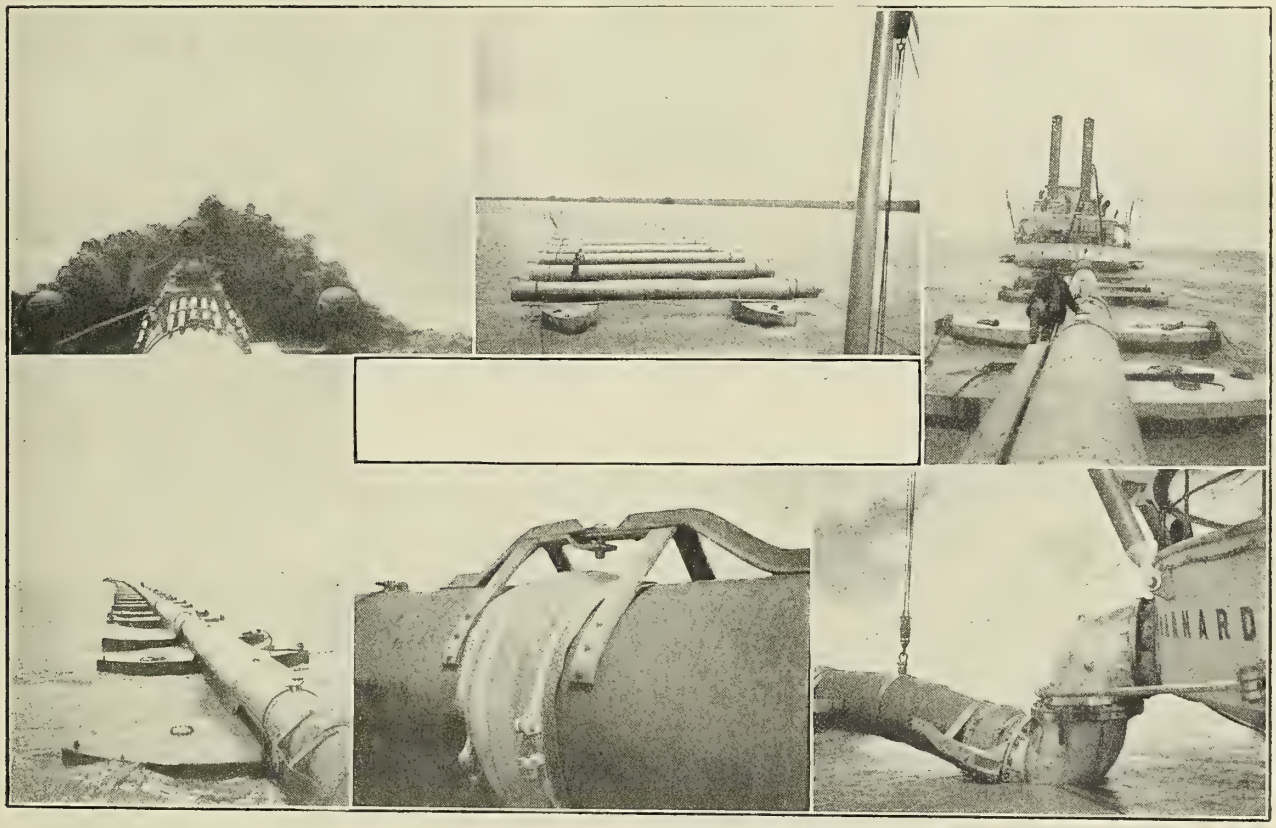
On the superstructure deck the forward end is occupied by the cabins of the officers. The suction well has a clear overhead space carried right up through the superstructure deck to the skylights above the cabin deck. This space is necessary for the suction hoisting tackles, which are suspended from an I-beam over the well, 27 feet above the water-line. Thus the cabins are built on either side of this opening, extending the length of the suction well. The cabins are all very large and airy. Accommodation is provided for twenty officers. The rooms at the after end are fitted up as lavatories, bath rooms, linen lockers, and stewards' store room.

Aft of this house the superstructure deck is cut by the skylight over the pumping engine. Aft of this again are two smaller houses abreast, the officers' mess room and stores on the starboard side, with the galley, pantry, and refrigerator on the port side. The galley deserves special mention for its size, excellent location, and complete equipment.

Aft of these two houses the smokestacks pass up through the deck. The last house on the superstructure deck is the crew's mess room. It has a room in the starboard end for cooks. There is no structure above the firemen's house on the main deck. It is used as a boat deck, and the four boats which form the equipment of the dredge are stowed there.

The pilot house is built over the after end of the main house on the superstructure deck. It is a handsome airy room 12 by 12 feet, and stands 40 feet above the water-line, commanding the whole situation in true Mississippi River style. From the pilot house communicating systems run to all the working platforms, so that the propelling engines, pumping machinery, suction operating gear, and hauling winches may all be worked to co-operate as desired.

It will be noticed that one of the features of the *Barnard* is the attention paid to the comfort of the crew. Not only are the quarters airy and comfortable for the warm weather of the Gulf, but the sanitary conditions are well provided for. Every window and door in the living quarters is provided with shutters and mosquito netting. This latter feature is of course especially important in malarial districts. All the cabins and living quarters



DISCHARGE AT END OF PONTOON LINE. VIEW OF PONTOON LINE FROM DREDGE. PONTOON LINE CONNECTED FOR TOWING. CONNECTION IN PONTOON LINE. VIEW OF DREDGE FROM PONTOON LINE. SWIVELING ELBOW CONNECTION TO DREDGE.

are supplied with running water, both hot and cold. The lavatories all have porcelain fittings, and the crew as well as the officers have porcelain bath tubs. The laundry adds greatly to the comfort of all on board, especially when stationed in isolated places. A complete steam-heating system is also installed for winter service up the river.

A more detailed description of the machinery follows:

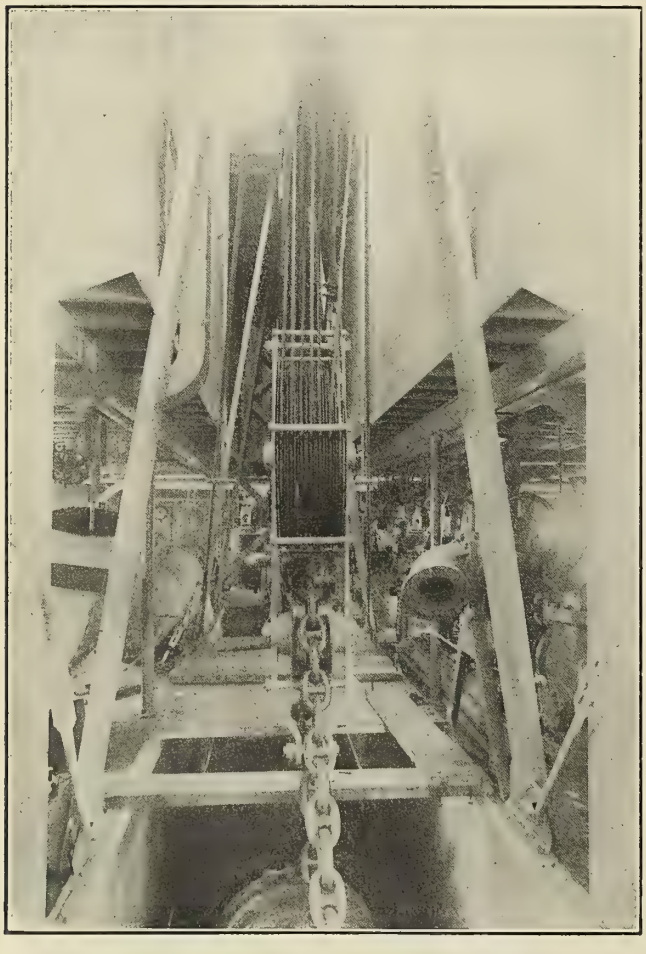
STEAM PLANT.

There are two main boilers of the Scotch double-ended marine type. They are 11 feet in diameter and 18 feet 2½ inches long, built for 170 pounds pressure, in conformance with the rules of the United States Supervising Inspectors. Each boiler has four corrugated furnaces, 43 inches inside diameter, opening into a common combustion chamber of unusually large dimensions. The total heating surface is 4,686 square feet, and the total grate surface 172 square feet, giving a ratio of 28.3 to 1. These boilers

U. S. DREDGE BARNARD ON THE DELAWARE RIVER.

have given great satisfaction under rather severe conditions; when dredging and cruising simultaneously they burn over 20 pounds of coal per square foot of grate surface per hour, and this under natural draft.

There are two main stacks, 52 inches in diameter, and 60 feet in height above the grate bars. The bunkers are situated along the sides of the ship abreast of the boiler room; they have a capacity of 120 tons. The auxiliary boiler is located on the main deck, over the starboard main boiler. It is of the Scotch, dry-combustion-chamber type, 5 feet in diameter and 5 feet long.

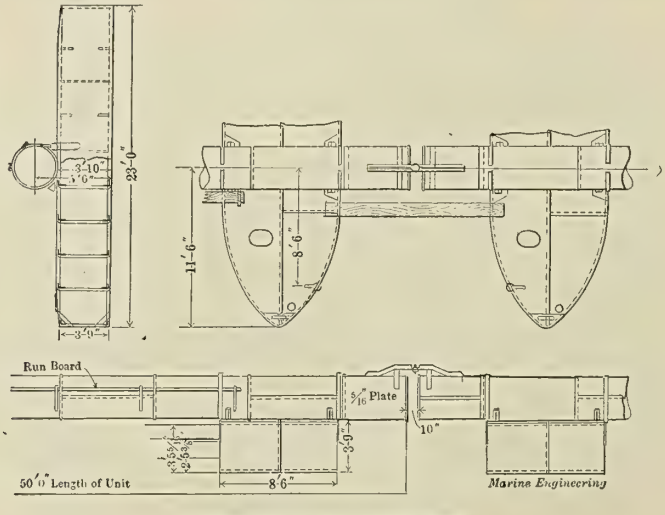


SUCTION WELL, LOOKING AFT, SHOWING TACKLE.

DREDGING MACHINERY.

The dredging pump and engine are direct connected, with axes athwartship. The pump room is located just abaft the suction well, and extends clear across the ship. Beginning at the port side come the L.P., I.P., and H.P. cylinders respectively, with the pump over on the starboard side of the ship. The dredging pump has a suction pipe 38 inches in diameter, which divides and enters at both sides of the casing. The discharge pipes are 36 inches in diameter, and the discharge leaves the pump at the top. The casing is in three sections bolted together. It shows a rectangular cross section through the axis; that is, the side discs of the casing are flat, and the outer sweep is a plain curve, terminating in a rectangular discharge pipe 32 by 32 inches.

The pump casing is fitted throughout with renewable liners, this being necessary owing to the gritty material handled. There are heavy cast-iron throat rings in the suction connections, which are also renewable. In the sides of the casing opposite the outer edge of the runner are recesses into which wearing rings are fitted. These wearing rings have chilled faces adjacent to corresponding rings on the runner. They are adjustable by means of bronze setting screws, so that as the clearance wears greater it may be reduced, and the pump thus kept always at its highest efficiency. The pump runner is of the enclosed type, 90 inches in diameter, made of cast steel, with wearing strips of chilled cast iron on its outer edges, corresponding to the adjustable wearing



PONTOON PIPE-LINE CONNECTIONS.

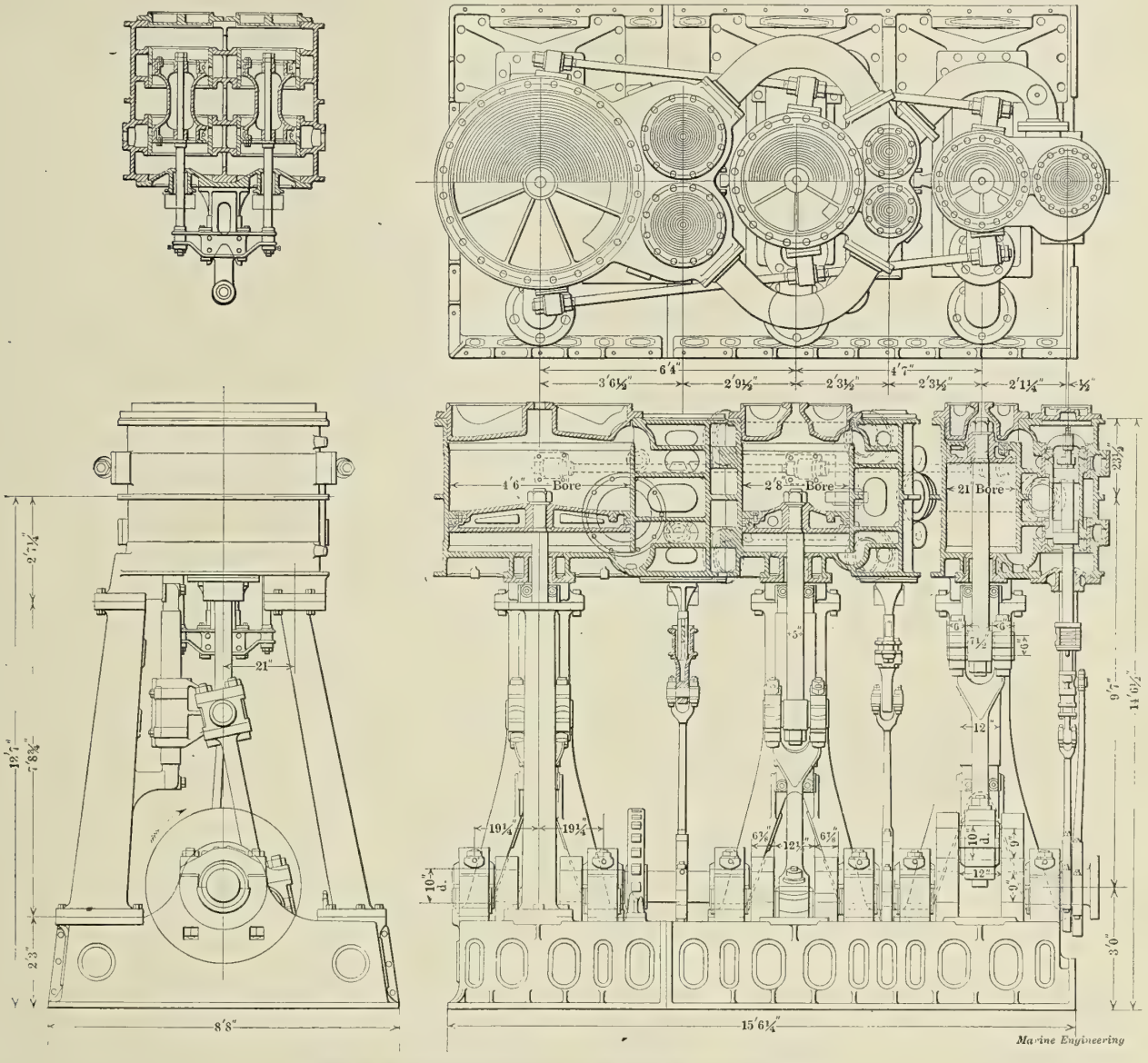
rings described in the casing. The runner is 32 inches wide across its face.

The pump shaft is of steel, 12 inches in diameter at the runner, and 10 inches in diameter at the ends. It runs out through the suction elbows through stuffing boxes bushed with bronze. The lower sections of the pump casings form an integral part of the bed plate, to which the engine bed plate is bolted. The whole structure, engine and pump, rests upon a substantial foundation, which comprises four keelsons in addition to the center keelson. There is almost no vibration apparent in the pump room.

The main dredging pump is driven by a vertical, inverted, triple-expansion engine; cylinders 21, 32, and 54 inches, with a stroke of 26 inches. This engine is of the modern marine type, with cast-iron back housings, and open cast-iron columns in front. The valves are all of the piston type, with two each for the I.P. and L.P. gears. The high-pressure valve is an interesting arrangement of the Meyer cut-off in a piston valve. It consists of one piston-valve within another. The valve stems are also concentric with each other, the cut-off valve stem being a sleeve fitted with stuffing box through which the main valve stem passes. But the valve stems are not concentric with the valves. This eccentricity of the valve stems is arranged in order to hold the sections of the cut-off valve against turning while adjustment is made by turning the valve stem. The whole arrangement is very neat and compact and has given great satisfaction.

The speed of the engine is controlled by a Pickering governor, acting directly on the throttle valve. It varies from 110 to 130 revolutions per minute, depending on the conditions of pumping. The power developed is approximately 1,000 indicated horsepower.

All working parts of the engine are of ample dimensions, with a complete water and oil service suited for continuous running. The journal service is supplied by a Blake horizontal duplex pump, 6 and 4 by 6 inches. In this engine room is also a sanitary and general service Blake horizontal duplex pump, 5¼ and 4¾ by 5 inches. The pumping engine has its own condenser and equipment of auxiliaries. The condenser has 2,400 square feet of cooling surface. The air pump is a Davidson valveless simplex, 9 and 18 by 10 inches. The circulating pump is the New York Shipbuilding Company's standard 10-inch size.



DESIGN OF TRIPLE-EXPANSION ENGINE DRIVING CENTRIFUGAL PUMP.

There are two different types of suction heads supplied for different kinds of material to be worked. The drag head has a straight cutting edge, 6 feet 6 inches wide, which scrapes up the material as it is dragged along by the motion of the dredge. The body is of cast steel and cast iron, with the cutting edge of steel plate. Bars are placed across the opening to prevent large solid bodies, such as logs of wood, etc., entering to damage the pump.

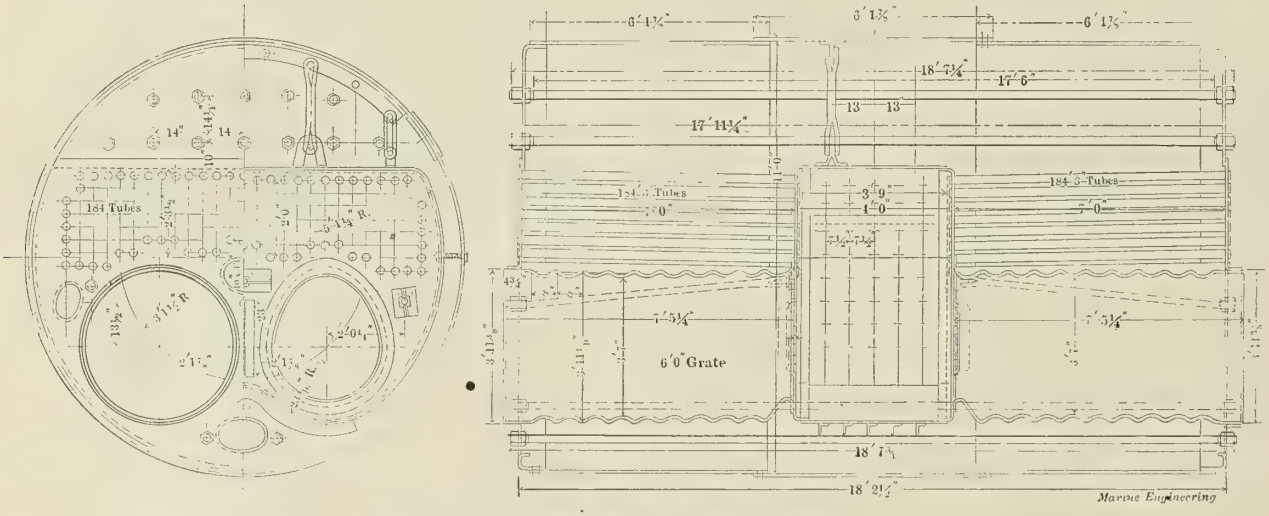
Near the cutting edge is a series of small holes for admitting water with the material to be dredged. There is also a large relief valve which may be opened to admit water, in the event of the mouth being choked entirely. This type of head is particularly suited to a sandy bottom.

The other head which may be used on the suction pipe is called the jet or suction sinking head. It is similar to an inverted scoop and is built up of steel plates with 8- by 1-inch screen bars across the opening.

From a header on top radiate eight pipes 2 inches in diameter. These are arranged around the lower edges of the head in such a way that water may be driven through them under pressure, and thus loosen up the material to be dredged. This device is very effective in sticky mud and blue clay. When dragged lightly over the bottom this head sucks mud from a path 12 feet or more in width. To supply water for these jet pipes a large compound duplex-plunger pump is located in the port side of the suction well at its forward end. It has steam cylinders 10 and 18 inches, water cylinder 14 inches, with a stroke of 16 inches, and can supply water up to 100 pounds pressure. This water is carried in an 8-inch hose along the main suction pipe to the header mentioned on top of the sinking head.

From the trunnion elbow at the forward end of the well the 38-inch suction pipe turns aft through another elbow and runs through the compartment on the starboard side of the well. On entering the pumping-engine room it couples onto a "Y" pipe which admits to both sides of the pump so as to counterbalance the thrust. The discharge, taken from the top of the pump casing, enters a switching valve, which is operated by a shaft extending up to the starting platform of the pumping engine. This valve can be locked in three positions, to discharge to starboard, to port, or equally on both sides. The starboard discharge goes directly overboard through a short connection. The port discharge also leads overboard through a pipe running across the ship; but by the removal of a section of this pipe an elbow may be inserted to connect with the astern discharge. The discharge from the port side of the switching valve then leads aft from the center of the pump room, between the boilers, through the propelling-engine room and after compartment, out through the stern of the ship. The discharge piping is all 36 inches in diameter.

On each bow of the dredge is a powerful hauling drum containing 2,000 feet of 1-inch wire cable. These drums are operated by double vertical engines, 8 by 8 inches, made by the American Ship Windlass Company. By means of these cables, attached to



LAYOUT OF ONE OF THE MAIN BOILERS ON BARNARD.

From the suction head the suction pipe leads to a trunnion elbow at the forward end of the well. The elbow passes through the starboard bulkhead of the well through the stuffing box in which it swivels. Opposite the stuffing box a projection on the elbow extends across the well to a bearing in the port bulkhead. These two bearings support the forward end of the suction pipe. The pipe is about 50 feet long and when up in a horizontal position it is clear of the water and folds up in the suction well somewhat like the blade of a jackknife in its handle.

The suction pipe is operated by means of two wire-rope tackles, one supporting it in the middle and the other at the suction head. The leads of these tackles run to the suction hoisting engine just forward of the well. They are arranged so that the pipe is kept nearly straight as it is raised and lowered about the trunnion elbow. In the middle of the pipe is a section of 38-inch rubber hose 15 feet long. This gives flexibility to the pipe to allow for lateral movements of the dredge, or uneven working of the hoisting tackles. The suction head can be lowered to dredge to a depth of 40 feet. The hoisting tackles are fixed to a 15-inch I-beam which runs along amidships over the well 20 feet above the main deck. It is supported by four "A" frames built over the suction well. The engines of the suction hoisting winch are 9 by 12 inches, and capable of handling the pipe instantaneously, even though half full of sand. Quick work is often necessary to prevent the suction head being caught under the ship in the event of a side current striking the dredge.

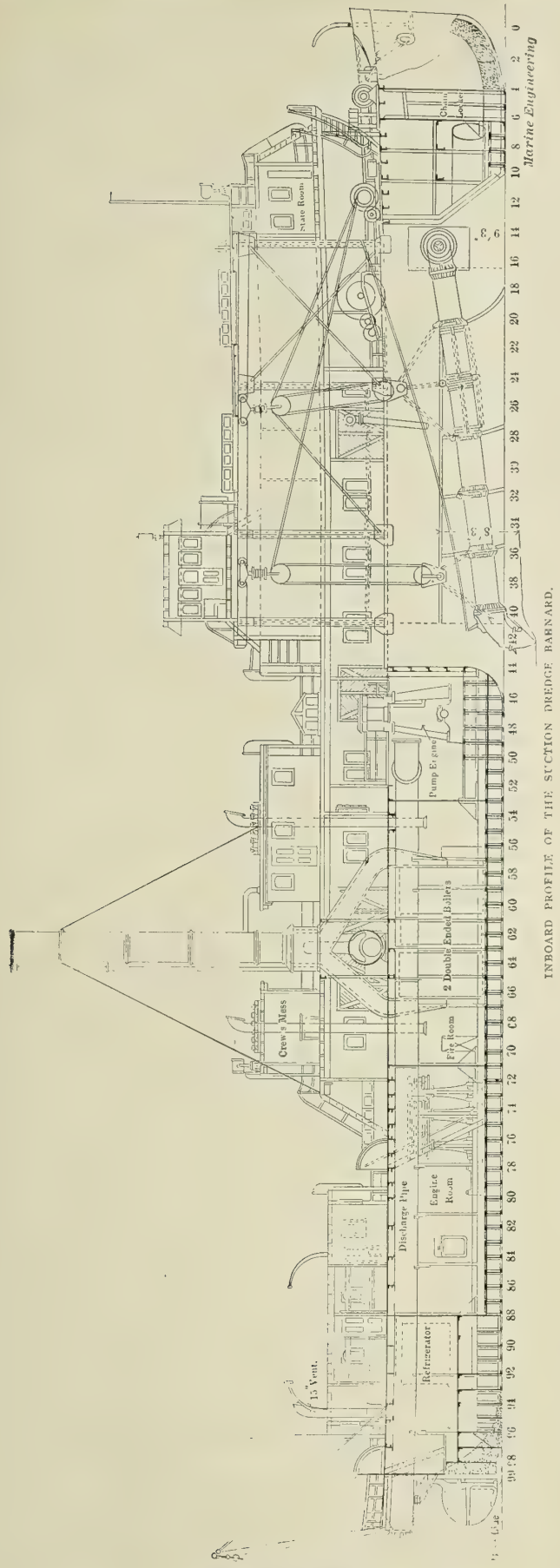
4,000-pound anchors, the dredge can be held in position or hauled ahead as desired.

The dredge has a complete installation of propelling machinery capable of giving her a speed of 10 knots. She has twin screws, with triple-expansion engines 12, 18, and 30 inches by 18 inches stroke. At 200 revolutions per minute they develop 900 indicated horsepower. The engines are of the regular marine type, with open front columns and piston valves throughout.

The propelling-engine room is unusually large and roomy. It extends for 34 feet near the after end of the ship, and occupies the entire width. There are no shaft alleys as the stern tubes start at the after engine-room bulkheads. The propeller shafts connect to the short section of line shaft by means of a Lovekin patent loose coupling. In the propelling-engine room is ample room for all auxiliaries, including evaporating plant and a two-ton refrigerating plant with its ice box.

The propelling-engine condenser has 1,400 square feet of cooling surface, with the usual outfit of auxiliaries, Davidson air pump, and 8-inch New York Shipbuilding Company's centrifugal circulating pump. The feed and filter tank is unusually large, to take care of any dirty water that might work in from the Mississippi. It is 12 feet long, 3 feet wide, 2 feet 6 inches deep. The filtering is done very effectually with rice straw.

There are three Blake vertical duplex feed pumps, 7½ and 5 by 6 inches, all located at the forward bulkhead of the engine room. A separate Blake horizontal duplex pump, 6 and 4 by 6 inches, is used to feed the auxiliary boiler.



INBOARD PROFILE OF THE SUCTION DREDGE BARNARD.

DECK MACHINERY.

The steam-steering engine is located on the main deck over the pump room. It is a Williamson, 5 by 5½ inches. The rudder has 50 square feet area. The rudder quadrant had to be kept below the discharge pipe which runs amidships out through the stern. The anchor windlass is of the pump brake type, built by the American Ship Windlass Company. There are four single barrel steam capstans, one at each corner of the ship. The engines, 5 by 8 inches, are placed below the main deck. They were also built by the American Ship Windlass Company.

ELECTRIC-LIGHTING PLANT.

The generating set is located on the main deck over the forward or pumping-engine room. It consists of a 9 by 6-inch Sturtevant vertical engine direct connected to a 15-kilowatt generator, 350 revolutions per minute. The switchboard was designed and built by the New York Shipbuilding Company. Besides 130 16-candle-power incandescent lights, in various parts of the ship, there are two 1,200-candle-power arc lights, and two 18-inch searchlight projectors. These searchlights have controlling gear for operation from various parts of the ship.

EVAPORATING AND DISTILLING PLANT.

The evaporator is located in the after end of the propelling-engine room. It is of the Reilly make, 2,500-gallons capacity per 24 hours, with a distiller of 500-gallons capacity for drinking water. The plant is supplied with the necessary feed, circulating, and fresh-water pumps. There are three cylindrical storage tanks for the fresh water distilled, two located in the forward peak and one in the hold on the port side of the suction well, with a total capacity of 5,400 gallons.

REFRIGERATING PLANT.

The ice machine is located in the port side of the propelling-engine room. It is built by the Remington Machine Company, of Wilmington, Del. The two ammonia compressing cylinders are single acting, 5 inches diameter, 9 inches stroke. The steam cylinder is 8 inches diameter by 8 inches stroke; all cylinders are vertical. The ammonia condenser is a vertical cylinder standing on a separate base. The ice-making tank has eighteen ice cans, holding 100 pounds each. A Deane horizontal duplex pump, 4½ and 3¾ by 4 inches, circulates the brine.

The cold-storage rooms are located aft of the engine-room bulkhead, and have a capacity of about 1,000 cubic feet. The acceptance test of the plant required that it should be run in 6- or 8-hour watches and with 6- or 8-hour stops, and make 600 pounds of ice per day; and at the same time maintain a temperature of 34 to 36 degrees in the cooling chambers. The entire refrigerating plant has ample capacity to fulfil these requirements under the severe conditions found in the Gulf of Mexico.

MACHINE SHOP.

In order to keep the equipment of the *Barnard* in repair, without recourse to machine shops up the river, she has a very complete machine shop of her own. It occupies the main deck on the port side of the suction well forward of the pumping-engine room.

The machines consist of:

One 24-inch screw-cutting lathe, with 16-foot bed, made by the New Haven Manufacturing Company, of New Haven, Conn.; one 25-inch drill press, made by the Fairbanks Company, of New York; one 18-inch crank shaper, made by the Ohio Machine Tool Company; one combined bolt and pipe threading machine, by the Jancke Manufacturing Company, of Erie, Pa.; one emery grinder with 12-inch wheels, made by the Diamond Machine Company, of Providence, R. I.

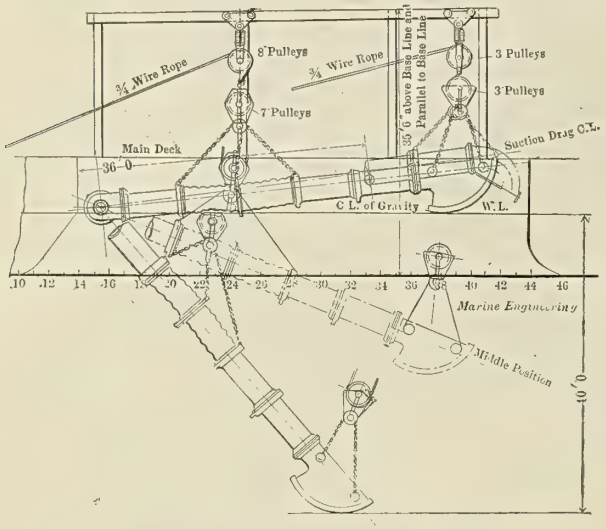
The machine-shop shafting is driven by a 7 by 9-inch vertical single engine, made by the New York Safety Power Company, of New York. All the machines have a very complete equipment of tools. There are also four hand vises and a well-selected set of hand tools, such as stocks, dies, reamers, files, etc.

The blacksmith and carpenter shops occupy the deck at the starboard side of the suction well. The blacksmith shop has a 42-inch circular forge with power-driven blower, made by the B. F. Sturtevant Company, of Boston, Mass.

PONTOON LINE.

The pontoon line consists of eighteen sections each 50 feet in length and 36 inches inside diameter. Each section is supported by two pontoons, one near each end. Each pontoon is an elliptical shaped float made of ¼-inch plate. They are 23 feet long, 8 feet 6 inches wide, 3 feet 9 inches deep, pointed at each end with arcs of 11 feet 3 inches radius. The pontoons are set athwart the pipe and secured by suitable saddles and bands.

The sections of the pontoon pipe are held together by heavy brackets riveted to the ends. These brackets connect the sections by means of heavy bronze pins and allow the sections to swing about each other in a horizontal plane. The end section of the



SHOWING FLEXIBILITY OF SUCTION PIPE.

pontoon pipe has a baffle plate 5 feet wide from the end of the pipe. This baffle plate is controlled by a windlass on the last pontoon, and may thus be set and held at any angle. This deflection of the discharge from the pipe reacts on the water and swings the end of the line in any direction desired. It must always be used unless the last section is secured; for if not, the reaction of the discharge plunging into the water would cause the pipe line to coil up on itself.

In order to make a continuous conduit for the material in the pipes, the spaces between the sections were connected by canvas sleeves, as shown in the illustration. These sleeves were made of 16-ply canvas firmly riveted and sewn together. The ends are slipped over the half-round finish on the pipe ends and then clamped. This arrangement is much cheaper than rubber and equally effective. It is the idea of Major Patrick, of the U. S. Engineer Corps.

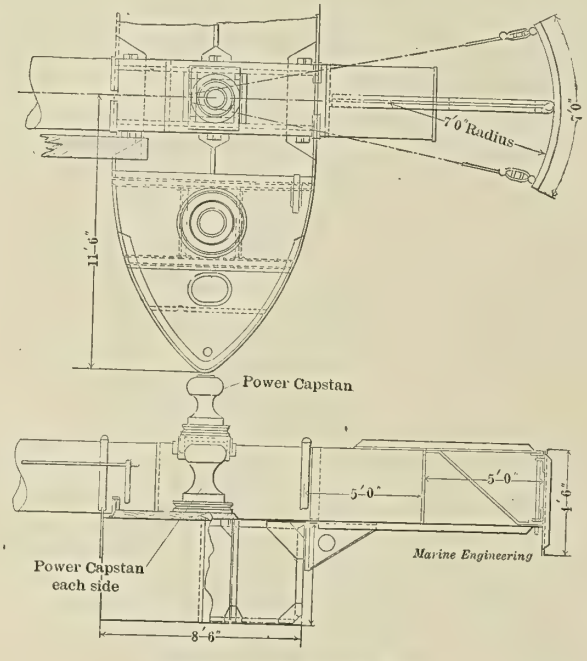
The pontoon line is connected to the dredge by means of a swivelling elbow, as shown. This elbow may be connected at the astern discharge, or at either of the side discharges. Shear legs over each discharge facilitate the handling of the heavy cast-steel elbows. The sponsons forward of the side discharges are for the protection of the discharge elbows when connected at the side.

OPERATION.

When dredging through the pontoon line the dredge is anchored and dropped back on the hauling cables. It can then be hauled ahead or stopped as desired. The jet-sinking head is used when the dredge is on the cables, and the method of operation is to let the suction head sink down about 18 feet in the mud, then hoist up and haul ahead to a new position. The suction pipe is then lowered again to its work. This method does not involve stopping the pumping engine, for the suction head is not raised out of the water. The result is a series of holes along the lead of the cables. A hole 12 feet wide, 20 feet long, and 18 feet deep, is dug by this method in 3½ minutes, including the time used in hauling ahead, etc., ready for another cut.

The pontoons being placed athwart the pipe they offer great resistance to towing when the pipe is coupled up for dredging. In fact, with the currents found in the Mississippi it is almost impossible for the dredge to make any headway against the stream with her own propellers. For towing purposes, therefore, the sections of the pontoon line are disconnected and brought up broadside to the current so that the pontoons may be towed end on. This arrangement of the line is shown in the illustration.

When working in a strong current over a bar near deep water, the *Barnard* dispenses with the pontoon line. She then cruises with her propelling engines and simply discharges at each side. With light mud this method is very effective, for the current carries the material out over the bar into the Gulf before it has time to settle. Even with heavy sand the material is moved seaward a considerable distance before it settles. Either type of suction head may be used while cruising. The jet-sinking head is very effective in the mud, while the drag head is more effective in sand. This method of stirring up the bars at the mouth of the Mississippi has been used for many years. On a wide, shallow bar, where a pontoon line is out of the question owing to rough water, etc., the dredging must be done in this manner till sufficient draft is obtained for a hopper dredge. The present bar at the mouth of the Southwest Pass has 10 feet of water over the crest, and the first problem for the *Barnard* is to obtain a channel 20 feet or more in depth. The hopper dredges can then take their



DISCHARGE END OF PIPE LINE.

loads and carry them out to sea. Later on the *Barnard* will build up the jetties and levees along the banks by discharging through the pontoon line. A special type of elevated discharge is being built by which the material can be thrown well out on the bank.

The dredge is also provided with an elevated side discharge by means of which she may pump directly into barges alongside. But the great capacity of the pump—over 400 cubic yards of water and sand per minute—makes it a difficult matter to supply barges rapidly enough.

R. S. Cooper has recently been appointed manager of the New York office of the Independent Pneumatic Tool Company, of Chicago, with headquarters at 170 Broadway, New York.

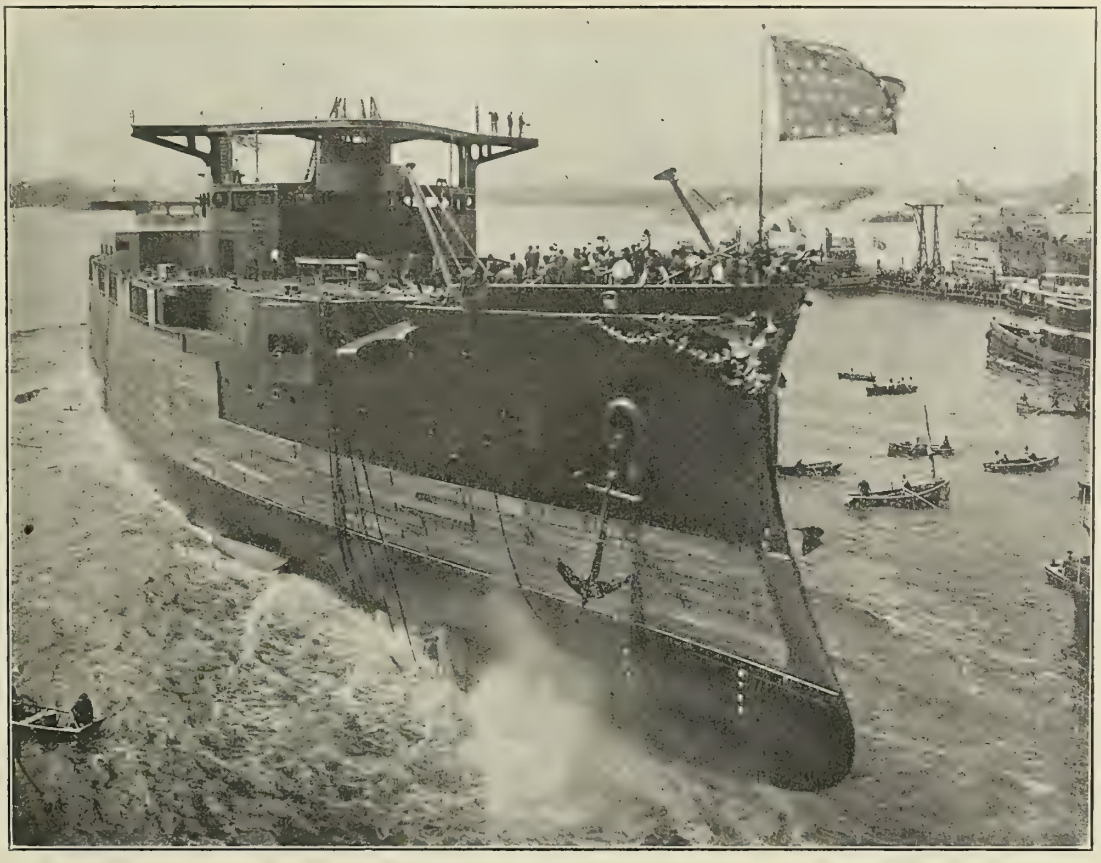
Edward P. Field, of the firm of Sadler, Perkins & Field, is recovering from an operation for appendicitis.

LAUNCHING OF TWO BATTLESHIPS.

On August 12 the United States battleship *Kansas*, one of the largest and most powerful yet laid down, was launched by Miss Anna Hoch, daughter of the governor of Kansas, from the yards of the New York Shipbuilding Company, Camden, N. J. The ceremony was performed with water, instead of the usual champagne.

On August 31, the new battleship *Vermont*, a sister of the *Kansas*, was successfully launched by Miss Jennie C. Bell, daughter of the governor of Vermont, at the yards of the Fore River Shipbuilding Company, Quincy, Mass.

The following description of the designs of the *Vermont* is taken from the official circular published for the information of bidders. It applies equally well to the *Kansas*.



THE LAUNCHING OF UNITED STATES BATTLESHIP KANSAS.

General dimensions and features of the vessel are as follows:

Length of load water-line	450 feet
Breadth, extreme, at load water-line	76 1/2 feet
Displacement on trial, not more than	16,000 tons
Mean draft to bottom of keel at trial displacement	24 1/2 feet
Gross draft, full load, about	26 3/4 feet
Total bunker capacity, coal, about	2,000 tons
Coal carried on trial	900 tons
Feed water carried on trial	66 tons
Trial speed at sea for four hours	18 knots

The hull is to be of steel throughout, fitted with docking and bilge keels.

Armament.—Main battery, four 12-inch breech-loading rifles; eight 8-inch breech-loading rifles; twelve 7-inch breech-loading rifles.

Secondary battery, twenty 3-inch, 14-pounder, rapid-fire guns; twelve 3-pounder semi-automatic guns; eight 1-pounder automatic guns; two 3-inch field pieces; two machine guns, caliber .30; six automatic guns, caliber .30.

The battery will be mounted as follows: The 12-inch guns in pairs in two electrically-controlled, balanced, elliptical turrets on the center line, one forward and one aft, each with an arc of fire of about 270 degrees. The 8-inch guns in pairs in four electrically controlled, balanced, elliptical turrets, two on each beam, at each end of the superstructure. The 7-inch guns in broadside on pedestal mounts on the gun deck behind 7-inch armor, each gun being isolated by splinter bulkheads of nickel steel from 1½ to 2 inches thick; forward and after guns arranged to fire right ahead and right astern respectively; other 7-inch guns to have the usual broadside train. The guns of the secondary battery in commanding positions, having a large arc of unobstructed fire, and protected wherever practicable. All the 7-inch guns are so arranged that their muzzles train inside the line of the side armor, thus leaving a clear and unobstructed side when it is desired to go alongside a pier or vessel. Arrangement will be made whereby the 3-inch guns on the main deck can be quickly and conveniently dismounted, housed, and secured.

Armor and Similar Protection.—The hull is protected at the water-line by a complete belt of armor 9 feet 3 inches wide, having a maximum thickness of 11 inches for about 200 feet amidships. Forward and aft of this the maximum thickness is 9 inches within the limits of magazines, from which points the thickness is gradually decreased to 4 inches at the stem and stern.

The lower casemate armor extends to the limits of the magazine spaces and reaches from the top of the water-line belt to the lower edge of the 7-inch gun ports on the main deck, and is 6 inches in thickness, the athwartship bulkheads at the ends of this casemate also being 6 inches thick. The casemate armor around the 7-inch guns on the gun deck is 7 inches thick, and the splinter bulkheads are from 1½ to 2 inches thick. The projection of 3-inch guns is nickel steel 2 inches thick. The upper casemate athwartship armor extending from the shell plating to the 12-inch barbettes is to be 7 inches thick throughout.

The 12-inch barbettes extend from the protective deck to about 4 feet above the main deck, and consist of 10 inches of armor in front and 7½ inches in the rear above the gun deck. Between the gun deck and protective deck there will be a uniform thickness of 6 inches. The barbettes will not have any special framing, the connection of the armor to the decks being sufficient. The 12-inch turrets will have a front plate 12 inches thick, rear plates 8 inches thick, and top plates 2½ inches thick.

The 8-inch barbettes will be 6 inches thick in front and 4 inches thick in rear, with the upper tube 3¾ inches thick and the lower tube 3 inches thick. The 8-inch turret front plate will be 6½ inches thick, the rear plate 6 inches thick, and the top plates 2 inches thick.

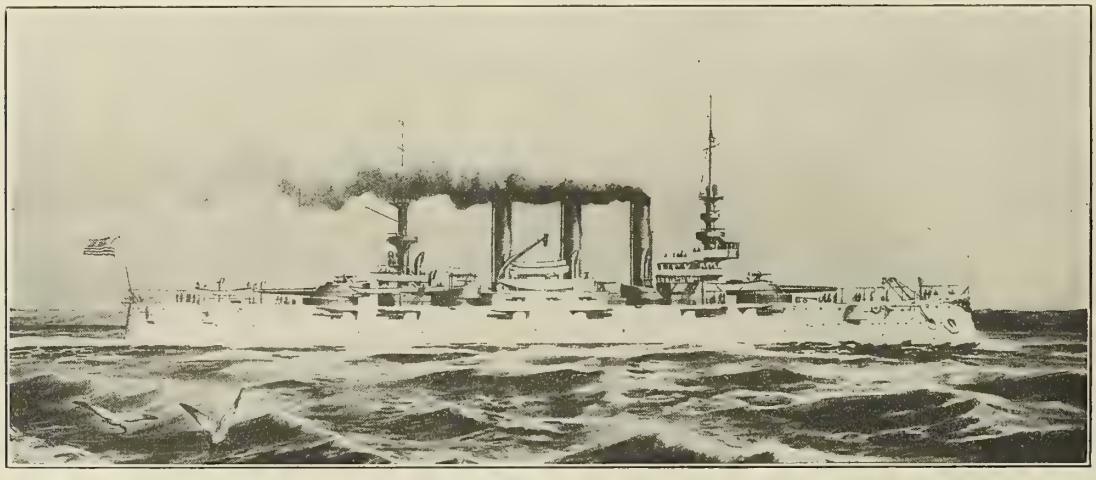
The conning tower and shield will each be 9 inches, and the signal tower 6 inches thick. An armor tube 36 inches in diameter will extend from the base of the conning tower to the protective deck, and will be 6 inches thick throughout.

Teak backing of a minimum thickness of 3 inches will be fitted behind all side, athwartship, and 12-inch armor; 2 inches of backing to be fitted behind the 8-inch turret armor; other armor will be fitted without backing.

ammunition rooms or ammunition passages to the deck on which it is required, or as near that as possible. These hoists will be driven at constant speed by an electric motor and will be arranged to deliver seven pieces per hoist per minute.

The number of hoists will be as follows: twelve for 7-inch, fourteen for 3-inch, 3-pounder and 1-pounder combined, and sufficient whip hoists to the tops. To supply the 7-inch hoists, four horizontal ammunition conveyors, operated by electric motors, will be fitted in ammunition passages for the transfer of ammunition from the handling rooms to the base of the hoists. The turret guns have regular turret ammunition hoists, operated by electric power, these hoists leading directly from the handling rooms or the ammunition passages to the turrets. For transporting 12-inch, 8-inch, and 7-inch ammunition, trolleys and tracks will be fitted in handling rooms, passages, and shell rooms.

Propelling Machinery.—The engines will be of the vertical, twin-screw, four-cylinder, triple-expansion type, of a combined I.H.P. of 16,500. The steam pressure will be 250 pounds. The stroke will be 4 feet. The ratio of high-pressure to low-pressure-cylinder will be about 1 to 7, the diameters being respectively 32½, 53, 61, and 61 inches, giving the required I.H.P. at about 120 revo-



GENERAL APPEARANCE OF VERMONT TYPE OF BATTLESHIP.

Protective Deck.—There is a complete protective deck extending from stem to stern, the deck being flat amidships, but sloped at the sides throughout, and sloped at each end. It will be built up of 20-pound plating throughout, with nickel steel of 40 pounds on the flat and of 100 pounds on the slopes.

The following nickel steel protection is also to be fitted: Hatch covers and gratings in the protective deck; splinter bulkheads on gun deck; sponsons and wing plates for two forward 3-inch guns on gun deck; bullet shields between wing plates for 3-inch guns and 7-inch guns; side protection and wing plates for 3-inch guns on main deck; turret shelf plates; conning-tower base plates; 7-inch gun port sill plates; 80-pound protection on ammunition-hoist trunks not otherwise protected by armor, and 80-pound protection on coaling trunks on slope of protective deck to the height of berth deck amidships.

Cofferdams about 30 inches thick and extending from protective deck level will be worked from end to end of the vessel, these cofferdams being extended above the berth deck, forward and abaft the transverse armor, to a height of about 36 inches. The cofferdams will be packed with cellulose or other approved water-excluding material.

Ammunition.—The magazines and shell rooms are so arranged that about one-half the total supply of ammunition will be carried at each end of the ship. Magazine bulkheads adjacent to heated compartments, such as fire rooms, engine rooms, and dynamo rooms, are arranged with air spaces. The ammunition for 7-inch and smaller guns will be conveyed by hoists directly from the

lutions per minute. Each engine will be located in a separate watertight compartment.

There will be twelve boilers of the Babcock and Wilcox type, placed in six watertight compartments. They will have about 1,100 square feet of grate and 46,750 square feet of heating surface, and must be able to furnish steam for the main engines and all necessary auxiliary machinery and other steam machinery throughout the ship with an average air pressure in the ash pits of not more than 1 inch of water.

There will be three funnels, each 100 feet high above the base line.

The following auxiliary steam machinery of approved make and design, in addition to that pertaining to the main engines and dependencies, is to be supplied: steering engine; windlass engine; ash-hoist engine for each fire room; forced-draft blowers; ice plant with a cooling effect of 3 tons of ice per twenty-four hours; evaporating plant to consist of not less than four units, having a total capacity of 16,500 gallons of fresh water per day; a distilling apparatus capable of condensing at least 10,000 gallons of water per day. The vessel to be heated by steam throughout.

The weight of all machinery and tools, stores, and spare parts will be about 1,500 tons.

The vessel will be lighted throughout by electricity. The electric plant will consist of eight 100-kilowatt steam-driven generating sets, all to be of 125-volts pressure at the terminals, disposed in two separate and independent dynamo rooms. There will be fitted all the usual means of interior communication, such as

telephones, voice pipes, call bells, buzzers, gongs and annunciators, engine and steering telegraphs, revolution and rudder indicators, heeling indicators, automatic fire alarm, warning signals, alarm signals, etc. The total weight of equipment outfit, complete, will not exceed 355.28 tons.

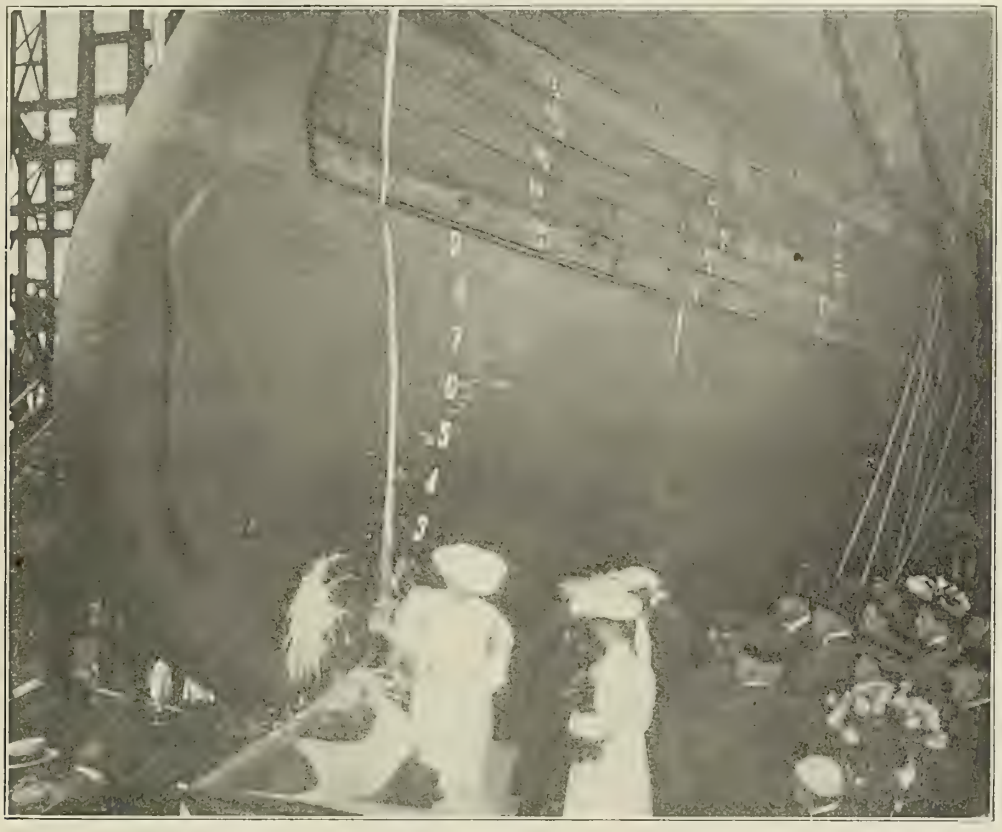
With the exception of the auxiliaries above mentioned to be operated by steam, all power on board of the vessel will be electric, as, for instance, boat cranes, deck winches, turret-turning motors, ventilation-blower motors, etc. All main compartments of the ship below the gun deck, except the coal bunkers, are to be provided with forced ventilation, there being no less than thirty-three blowers, with a combined capacity of 104,000 cubic feet per minute. Special attention has been given to spaces subject to habitually high temperature, such as engine rooms, fire rooms, and dynamo rooms. The ventilation system will be designed to cut the minimum number of watertight bulkheads. All blowers, except forced-draft blowers, are to be electrically operated.

The coal bunkers are to be arranged with satisfactory reference to the rapid and efficient supply of coal to the fire rooms, and have a maximum capacity of about 2,200 tons. There are to be provided for coaling not less than six winches, twelve booms, and all necessary fixed chutes, scuttles, hatches, and other openings. There will be two billboards and four hawse pipes; all hawse pipes to be so designed that stockless anchors may be stowed in them. There will be four heavy anchors, two of the navy type and two of the stockless type.

The following boats will be carried, adequate provision being made for their convenient stowage and handling by two electrically-operated boat cranes, eight pairs of boat davits, adjustable boat chocks, and all necessary fittings provided for this purpose: One 50-foot steam cutter; two 36-foot steam cutters; one 36-foot steam launch; three 33-foot launches; five 30-foot cutters; two 30-foot whaleboats; one 30-foot gig whaleboat; one 30-foot barge; two 20-foot dingeys; one 16-foot dingey; one 14-foot dingey; and two life-rafts.

The vessel is designed as a flagship, and the arrangement of quarters provides ample accommodations for the following complement: a flag officer; a chief of staff; nineteen ward-room officers; ten junior officers; ten warrant officers; and not less than 761 men, including 60 marines.

There is a lower bridge both forward and aft, and a flying bridge forward, according to the latest practice. On the flying bridge is fitted a screen of bronze for the protection of the men at the wheel, and there is also a bronze chart house forward. There are steel masts forward and aft, the foremast having an upper and lower top, the main mast a lower top only. Masts are arranged for wireless telegraphy. There is one signal yard on each mast, also a searchlight platform forward and aft, with a crow's nest on the foremast.



BREAKING THE BOTTLE ON THE BOW OF THE VERMONT.

A summary of the weights to be carried on trial is as follows:

	TONS.
Guns, mounts, magazine equipments, etc.	944.94
Ammunition, two-thirds full supply	393.97
Steam engineering, complete, with water in boilers, condensers, piping, etc., and stores, etc., not to exceed	1,500.
Reserve fresh water for steaming purposes	66.
Coal, normal supply	900.
Boats and outfits	51.27
Masts and spars	30.08
Equipment, complete, including anchors, chains, electric plant, etc., and two-thirds equipment stores	355.28
Miscellaneous stores and water, two-thirds full supply	81.32
Provisions, clothing and small stores, two-thirds full supply	146.84
Officers, crew, and effects	103.32
Total protection, including armor, armor backing, armor bolts, cellulose, and splinter bulkheads	3,992.

This leaves, as the weight of the hull, fittings, and protective deck, about 7,434 tons.

The percentage of completion of the *Kansas* is about 62, and of the *Vermont* about 65.

Marine Engineering

Published Monthly by

MARINE ENGINEERING

INCORPORATED.

17 Battery Place, - - NEW YORK

12 St. Benet Place, Gracechurch St., LONDON, E. C.

H. L. ALDRICH, President and Treasurer

PROF. W. F. DURAND, Advisory Editor

SIDNEY GRAVES KOON, Editor

GEORGE SLATE,
Vice-President and Advertising Representative

Branch Offices. { Philadelphia, Pa., Machinery Dept., The Bourse, S. W. Anness.
Boston, Mass., 170 Summer St., S. I. Carpenter.

TERMS OF SUBSCRIPTION.

	Per Year.	Per Copy.
United States, Canada and Mexico	$2.00	20 cents
Other countries in Postal Union	10/6	1/-

Entered at New York Post Office as second-class matter.

Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

The edition of the October issue of Marine Engineering comprises 5,500 *copies. We have no free list, accept no return copies and issue only enough to supply the regular demand, so that the* 5,500 *copies represent legitimate circulation.*

Engineers in the Navy.

The report of the Board of Inquiry into the *Bennington* disaster has been submitted to the Navy Department and has, in the main, been approved. The department has, however, gone further than its recommendations by ordering not only the court martial of Ensign Wade, who was in charge of the machinery of the *Bennington* at the time of the accident, but also that of Commander Young, who had charge of the ship itself. The latter officer has declared his expectation of being completely vindicated by virtue of the fact that he had, on numerous occasions, reported the boilers of the *Bennington* to be absolutely unsafe, and that, as his warnings were unheeded, responsibility for a tragedy which cost the Navy more than three times as many lives as were lost in action in the war with Spain rests primarily upon the department.

It seems to us that the only possible culpability of Commander Young might lie in the possibility that certain disagreeable rumors regarding the effect of San Diego rum upon the men immediately in charge of the boilers might have some foundation in fact, and have indicated a laxity in discipline, not, perhaps, without precedent in the Navy, but certainly without the possibility of justification. It is almost inconceivable, as one of our contemporaries remarks, that men in a sober condition, who were in possession of any knowledge whatever upon the subject of boilers, should have gone on firing under the boiler when the lack of pressure in the gauge could not but have indicated that something was wrong. It is also very much of a mystery how the air cock, not being closed, failed to sound a warning.

Whatever the true explanation may be, it is quite certain that those immediately responsible, as well as those ultimately responsible for the condition of affairs which made possible the explosion, are entirely beyond the reach of any human court martial; and that we must look for an improvement in matters to an entire rehabilitation.

This is the finding of the court:

"That the *Bennington* arrived in the harbor of San Diego, Cal., on July 19, 1905, from Honolulu, Hawaii, being *en route* to Panama to relieve the *Princeton;* that the ship was in an excellent state of discipline and in a good and efficient condition, with the exception of her boilers, which were in fair condition and efficient considering their age (about fourteen years) and the use to which they had been subjected.

"That on July 20, orders were received to proceed at once to Port Harford, Cal., and convoy the *Wyoming* to the navy yard at Mare Island, Cal.; that at this time the ship was coaling and her boilers and engines were being overhauled preparatory to her voyage to Panama; that upon the reception of telegraphic orders boilers 'A' and 'B' were ordered to be filled with fresh water and steam was ordered for 10 o'clock the following morning, July 21, 1905; that owing to the delay in obtaining the fresh water it was not practicable to commence filling the boilers 'A' and 'B' until about 5.30 o'clock A.M. July 21, and at about 8 o'clock A.M. they had been filled to high steaming level.

"That at about twenty minutes after 9 o'clock A.M. the steam gauge on boiler 'B' showed about 5 pounds of steam pressure, and at this time oiler Frank De Courtani, acting as water tender, directed D. N. Holland, fireman, second class, to close the air cock on boiler 'B'; that the said Holland climbed up and closed a valve, and almost immediately the steam gauge on boiler 'B' failed to register any pressure; that this was apparently not noticed by either the water tender or the fireman, and no attention appears to have been paid to the fact that the steam gauge failed to register, but they kept on working the fires and firing heavily; that when the steam gauge on boiler 'A' showed 135 pounds there was no pressure showing on the steam gauge of boiler 'B.'

"That at about a quarter to 10 o'clock A.M., the engines were turned over, using steam from boilers 'C' and 'D'; that as it was not thought that steam would be ready in boilers 'A' and 'B' before early in the afternoon, it had been decided to get under way and leave the harbor under boilers 'C' and 'D,' but steam appears to have formed much more rapidly that it was thought possible, and boiler 'A' was connected with boilers 'C' and 'D' at about twenty minutes after 10 o'clock A.M.; that no pressure was showing on the steam gauge of boiler 'B' at this time.

"That about this time a small leak developed in No. 1 furnace of boiler 'B,' and coal-passer A. J. Worthen was sent on deck by De Courtani, the acting water tender, to inform the boilermaker about the leak and request him to come below and attend to the same; and just about this time, as Worthen was leaving Dustin, the boilermaker, who was, we believe, on the berth deck, the explosion occurred.

"That no one seems to have noticed any escape of steam from the safety valves of any of the boilers, and no one can state that any of the safety valves blew off at any time that morning.

"That we can find no record of the safety valve of boiler 'B' having been overhauled since July, 1904, nor any positive evidence of it having been done, though orders had been given for this to be done in March, 1905; that there is no record of the sentinel valves having been overhauled since July, 1904; that the safety valves were set at 145 pounds, but *en route* from Honolulu to this port orders were given to carry the steam pressure at from 130 to 135 pounds, and not to exceed the latter, but the safety valves were not changed; that this order had been clearly understood; that the hand gear for lifting the safety valves was not in working order, and there is no record or direct evidence that the safety valves had been tested in accordance with the navy regulations."

Following the statement of its findings, the Court of Inquiry gives this opinion:

"The court is of the opinion that the explosion was caused by excessive steam pressure in boiler 'B,' which came about first by shutting the valve connecting the boiler with the steam gauge, instead of the valve on the air cock alone, as was intended; so that the steam gauge did not indicate the pressure in the boiler; second, by unusual and heavy firing in the boiler to get up a pressure which the gauge failed to show; third, by the failure of the sentinel and safety valves to lift at the pressure for which they were set, and the pressure increased without relief until it was beyond the strength of the boiler, which gave way in its weakest part, afterward found to be the corrugated flue of No. 2, the lowest or middle furnace, which collapsed."

As we have repeatedly stated in discussing not only this particular disaster but other features in connection with engineering in the United States Navy, the present is above all things an age of specialists; and what the Navy needs for the work connected with engineering duty on its ships is, primarily, a corps of specialists, whose entire training has tended toward the one end of fitting them for the duties which, in the service to which they are accredited, they are called upon to perform. The requirements of that service are continually increasing, as is evidenced by the fact that the latest ships of any size in the Navy have machinery plants of from 16,000 to 25,000 horsepower and upward, as compared with the plants ten years ago, which exceeded in only a very few instances 10,000 horsepower. As a direct effect of this, any change in the personnel should be toward a decided increase in every respect, not only in numbers, but also, and in perhaps larger degree, in general efficiency as well.

It would be idle to look to the naval academy for this. The situation demands instant relief, and no man leaving the academy could in less than five years from his date of graduation be safely entrusted with such responsibilities as are now lying around loose and uncared for—sad experience has proved that—and those five years must needs be spent in earnest and diligent work, actually in an engine room. It is safe to say that, were the entire equipment of the academy devoted to this sole purpose, at least ten years would elapse before we had an engineering force commensurate with our requirements—both quality and quantity considered. The only solution suggested, which appears to us to bear the hall-mark of sanity, is to open the door for the introduction of men from civil life. Not only this, but it will be found necessary to insure them treatment due a white man—not that usually accorded a yellow dog. Men possessing the requisite qualifications are men of standing and character. No other will do. This fact must be recognized. And such men will not enter the service unless they can be assured as considerate treatment as their training (in our great technical schools), their abilities, and their characters would provide them in their usual vocations ashore.

It is not at all necessary that the Navy should return to anything of the same general nature as the old engineer corps, which was abolished by the Personnel Law of 1899. We do not want a return to conditions in which the bitter feelings (on account of social matters and questions of rank and priority) between line officers on the one hand, and engineer officers of equal training, gentlemanliness, and capacity, on the other, furnished a beautiful target for cartoonists and professional humorists. The case is one where military titles, fuss and feathers, and gold lace, should not be allowed to dictate policy. The respect and dignity which the position of engineer on one of our large naval vessels should command can be attained only by granting to that officer privileges and authority commensurate with the responsibility which he is called upon to assume; and until the department and Congress look this matter squarely in the face and realize the truism of the above statement we cannot expect much relief. It is beyond all question that we *must* have a competent engineering force, and that in order to obtain it we must sacrifice, if necessary, all pet prejudices entertained by junior officers of the line in favor of their intrinsic, indisputable, and immense superiority over men whose duties require them to take care of machinery, in the proper operation of which they are liable to get their hands dirty. The general snobbishness responsible for the conditions which brought about the law of 1899 is, in many respects, attributable to the fact that officers of both the Army and the Navy have in many cases, particularly when they are quite young, come to look upon themselves as beings quite removed from ordinary personages, and to assume that the gold buttons which they wear entitle them to the universal obeisance of mankind.

ABSTRACT FROM RULES AND REGULATIONS OF THE AMERICAN BUREAU OF SHIPPING FOR SOUND, LAKE, BAY, AND RIVER STEAMERS, YACHTS AND TUGS.

EDITION OF 1904.

ARRANGED BY HARRISON S. TAFT.

(*Continued.*)

23. Riveting.

1. The style of riveting used in the longitudinal seams of the shell plating forward and aft is to be of the same type as that of said seams amidships.

2. With plates of different thickness the thicker of the two is to determine the size of the rivets used.

3. Lapped butts are to have the same style of riveting as if butt straps were adopted.

4. All double-riveting to be chain fashion, except for Bar Keels, Stems, and Stern Posts, where zigzag riveting is to be used.

5. Rudder rivets are to be of the same diameter as those used in the lower edge of the main sheer strake.

6. Bar keels, when 5 inches or more in depth, are to have a double zigzag riveted connection to the garboards.

7. Bar keels, stem and stern-post rivets are to be 1-4 larger in diameter than those used in the upper lap of the garboards.

8. The doubling clips to the frames at the margin plate in double-bottom vessels are to have 3 rivets in each flange on each side of said margin plate.

9. All angle-bosom clips at the reverse-bar butts are to have 2 rivets on each side of said butt. Those at all keelson and stringer angle-bar butts are to have 3 rivets on each side of said butts.

10. All angle clips on the frames and floors in the way of the keelson and side stringer angles are to have a 3-rivet connection to said floors and frames.

11. The Following to be Single Riveted:

(*a*) Longitudinal seams of all shell plating, except the garboards and the lower seam of the main sheer strake, when the weight of midship plating does not exceed 16 pounds.

(*b*) Upper lap of garboards, and the lower longitudinal seam of the main sheer strake, when the weight of the midship plates of said strakes is under 16 pounds.

(*c*) Upper edge of all main sheer strakes.

(*d*) Fore-and-aft seam strips of flush shell plating, when the weight of the midship plates does not exceed 22 1-2 pounds.

(*e*) All butts of the shell plating; of the deck stringers and deck plating; of all water-ballast tanks and inner-bottom plating; when said plating is under 7 1-2 pounds.

(*f*) Bulwark butts in all cases.

12. The Following to be Double Riveted:

(*a*) All floor-plate butts, when tonnage does not exceed 500 tons.

(*b*) Diagonal tie-plate connections to deck stringer plates.

(*c*) Butts of the deck plating, except in bulk freight vessels of the Great Lakes.

(*d*) Lower strake of the centerboard-trunk plating, when flanged to a flat plate keel.

(*e*) All flat-plate keel connections to adjoining strakes.

(*f*) Longitudinal seams of all shell plating, except the garboards and the lower seam of the main sheer strake, when the weight of midship plating exceeds 16 pounds.

(*g*) Upper lap of the garboards and the lower longitudinal seam of the main sheer strake, when the weight of the midship plates of said strakes is 16 pounds or above.

(*h*) All butts of the shell plating (except Bulwarks); of the deck stringers and deck plating; of all water-ballast tanks; and inner-bottom plating; to be at least double riveted, when said plating is of 7 1-2 pounds or more in weight.

(*i*) Hurricane-deck stringer plates and all outside plating in way of said decks above the main sheer strake.

(*j*) Fore-and-aft seam strips of flush shell plating, when the weight of the midship plates is over 22 1-2 pounds.

TABLE 9.—RIVETING SCHEDULE FOR SHELL PLATING.

Type of Vessel.	Sailing Vessels.		Vessels of the Great Lakes.			
			Bulk Freight.		Engines Fitted Aft.	
Tonnage.	500 under 1,000.	1,000 and above.	Under 2,000.	2,000 and above.	1,500 and under.	Over 1,500.
Butt straps of:						
Sheer strake	T R — $\frac{1}{2}$ L	T R — $\frac{1}{2}$ L	T R — $\frac{3}{5}$ L	T R T	[a] T R — $\frac{3}{4}$ L	T R T
First bilge strake	T R — $\frac{1}{2}$ L	T R — $\frac{1}{2}$ L	T R — L	T R T	[a] T R — $\frac{3}{4}$ L	T R T
Second " "	.....	T R — $\frac{1}{2}$ L	T R — L	T R T	[a] T R — $\frac{3}{4}$ L	T R T
M D stringer	.....	T R — $\frac{1}{2}$ L	.....	.....	.....	.....

[a] Measured from stern post.
T R = treble riveted. T R T = treble riveted throughout.
— $\frac{3}{5}$ L = for $\frac{3}{5}$ length amidships.

13. The Following to be Treble Riveted:

(*a*) All floor-plate butts, when tonnage exceeds 500 tons.

(*b*) All butts of flat-plate keels.

(*c*) All butts of continuous plate keels and keelsons.

(*d*) All butts of middle-line keelsons and side longitudinals.

(*e*) All butts of shell plates 54 inches or more in width.

(*f*) Butts of the deck plating in bulk freight vessels of the Great Lakes.

(*g*) All butts of the shell plating for 3-4 vessel's length forward from the stern post in vessels upon the Great Lakes with machinery located aft.

14. Countersinks should be in depth not less than 3-4 the thickness of the plate or the angle-bar flange.

TABLE 10.—RIVETING OF INNER BOTTOM PLATING.

Tonnage.	Under 2,000.	2,000 under 3,000.	3 000 and above.
Edges of the middle line strake.....	Single.	Double.	Double.
Edges of strakes adjoining the middle line strake.....	"	Single.	"
Edges of all remaining strakes.....	"	"	Single.
All butts of said plating to be double riveted throughout.			
IN VESSELS UPON THE GREAT LAKES WITH MACHINERY LOCATED AFT.			
All butts of said plating to be treble riveted for ½ length forward of engine room.			
When built for transporting of heavy dead weights	Over 2,000, under 3,000 tons. T R — ½ L amidships.		3,000 tons or above. T R T

T R = treble riveted. T R T = treble riveted throughout. — ½ L = for ½ length amidships.

24. Double Bottoms.

1. Double bottoms may be of the following types:

(*a*) Longitudinal girders on top of floors.

(*b*) Continuous side longitudinal plate girders extending from the shell to the inner-bottom plating.

(*c*) Continuous floor plates with intercostal side girders.

2. Longitudinal Girders on Top of Floors. Said girders are to be spaced not over 42 inches apart. They are to be of the same weight as the floor plates; with a continuous angle bar, equal to the reverse bars in size but to the end frames in weight, worked on the upper and lower edges for securing said girders to the inner-bottom plating and to the reverse bars with their reverse clips. In lieu of the reverse clips vertical stiffeners may be worked, extending from the top of the girder plate to the lower side of the floors. Said girders are to be worked continuously through all watertight floors.

3. Continuous Side Longitudinals.

(*a*) The plating of said longitudinals need not exceed 22 pounds. Said longitudinals are to be secured to the inner-bottom plating with a continuous angle bar of reverse-bar flange size, but equal to the end frames in weight. Bracket plates of the same weight as the girder plates are to be worked at alternate frames at top and bottom on each side of the girder plates; being secured to the inner bottom plating and longitudinals with angle clips of the same scantling as the bars on top of said longitudinals. The clips connecting said brackets to the center-line girder shall be worked double.

(*b*) When the tonnage is 1,000 and above, solid manhole intercostal floor plates of the same weight as the side longitudinals are to be fitted on every eighth frame; having double clips of reverse-bar size to the longitudinal girders, but a single clip of the same scantling as the longitudinal top bars to the inner-bottom plating.

4. Solid Floors.—When double bottoms are built with solid floors, said floors are to extend from the shell to the inner-bottom plating, and from the center line to the margin plate. They may be 2 1-2 pounds less in weight than ordinary floors, except at bulkheads and under the engine bed, where full weight must be maintained, with double reverse bars on top. An intercostal-plate girder of the same thickness as the inner-bottom plating is to be worked midway between the center line and the margin plate; being clipped to floors, shell, and inner-bottom plating with angle clips equal to reverse bars.

5. All lower bilge keelsons may be omitted in way of all inner bottoms.

6. When double bottoms are not worked continuously the usual bilge and other keelsons must be worked throughout the spaces where no double bottom is worked. Such keelsons are to lap the inner-bottom girders, so as to maintain the longitudinal strength.

7. The margin plates may be flanged to the shell plating, or be secured to same with a continuous angle bar (the frame being cut), or by watertight intercostal staples worked around the frames and reverse bars.

8. When the frames are cut, bracket plates, equal to the weight of the margin plates, are to be worked on both sides of the margin plate at every frame. Said brackets shall have a four-rivet connection to the frames and be secured to the margin plates by angle clips equal to reverse bars, with at least four rivets in each flange.

9. When the inner-bottom plating is not worked through the watertight bulkheads at the ends of the machinery space, said plating is to be connected to said bulkheads with a single bar equal to the reverse bars.

10. All inner-bottom plating, except the margin plates and the plating in the machinery space, is to be of the same weight as that of the lower half of watertight bulkheads. It shall be worked fore-and-aft in length equal

to the shell plating. The margin plates and all the plates in the machinery space, except those under the engine bed, are to be 2 1-2 pounds heavier than said bulkhead plating, if said bulkhead thickness is under 17 pounds. When said bulkhead plating is 17 pounds or above, the inner-bottom plates throughout are to be of the same weight as the bulkhead plating. When the weight of the inner-bottom plating is and above 15 pounds, said plating fore-and-aft of the 1-2 length amidships can be reduced by 2 1-2 pounds. The plating under the engine bed is to be of approved thickness.

11. Manholes may be cut in all non-watertight floors, longitudinal girders, and inner-bottom plating to allow of ready access to all parts of the double bottom.

12. Solid watertight floors with double frame bars are to be fitted under all watertight bulkheads. Floors under the engine bed are to be of the solid manhole-floor type, and shall be fitted at every frame.

13. In engine and boiler rooms of vessels of 1,000 tons and above, solid intercostal floor plates are to be fitted on every third frame. A solid floor must be located under all boiler bearings.

14. A center line watertight keelson should be worked in every case; secured by double angle bars to the inner-bottom plating.

15. Fore-and-Aft Peaks. A watertight deck or flat of the same weight as the lower half of watertight bulkheads should be worked on the lower or main deck beams when it is intended to carry water ballast in either the fore or aft peak. Auxiliary beams should be fitted between the regular beams under said plating, with knee brackets equal to the weight of the adjoining deck stringer, with at least a four-rivet connection to both the frames and the beams.

16. All watertight ballast tanks or compartments, except the fore-and-aft peaks, which extend to the main or lower decks, are to have a center-line bulkhead for the entire depth of said tank. Said bulkhead is to equal top plating of tanks and be well stiffened, similar to all other bulkheads.

17. All water ballast and inner-bottom compartments must be tested with a head of water extending at least two feet above the deep-load water-line.

25. Hatchways and Deck Openings.

1. Coaming plates at all deck openings should extend to the lower side of the deck beams to which they are secured; being equal to the thickness of said beams. All coaming plates of freight hatches and deck houses should have rounded corners. The freight hatches on a weather deck are to have their coaming plates standing at least 12 inches above the deck planking with a half-round beading on the upper edge. Suitable iron supports are to be fitted on the inside for carrying the hatch covers.

2. When a deck house is fitted over an engine or boiler opening on the weather deck the coaming is to stand at least 12 inches above the deck planking; but 30 inches if no deck house is fitted over said openings. In the latter case a half-round beading is to be worked on the outside of the top edge of the coaming, with a suitable support on the inside of the coaming around the fire room hatch so as to carry a metal grating.

3. There shall be an angle bar equal to the main frames, securing the coaming plates to the steel deck or tie plates, extending all around the hatch; the upper edge of the flange which is secured to the coaming plate being caulked watertight.

4. When freight hatches are over 12 feet in length or width, the coaming plates are to be 25 percent thicker than the beams to which they are secured; and are to stand at least 16 inches above the deck planking. The tie plates at the sides of the hatches are to be double their tabulated width for one beam space beyond the ends of the hatch opening. If a steel deck is worked the strake of plating next to said coamings is to be 2 1-2 pounds heavier than otherwise for same distance.

TABLE II.—HATCH BEAMS AND CARLINGS.

Hatch Length in Feet	a. Hatch Beams.	Hatch Width in Feet.	c. Hatch Carlings.
Over 16 under 22.....	b. One web plate shifting beam.	10 and under.	One.
22 not over 30.....	b. Two " " " "	Over 10 not over 15.	Three.
30 and above.....	Plans to be submitted.	Over 15.	Plans to be submitted.

a. Besides the above increase in deck ties or deck plating.
b. Of same weight as coaming plates; in depth at least ⅔ whole depth of said coaming plates; fitted with double top and bottom angle bars equal to reverse frames; bolted to double clips on side coamings; arranged so as to support fore and aft wooden carlings.
c. Supports for same, with a fore-and-aft bearing surface of at least two inches deep to be riveted to end coamings.

5. Wooden hatch covers are to equal the thickness of the main deck planking.

6. In vessels on the Great Lakes with extreme wide hatches and low coamings, the side coaming plates should extend below the deck beams a distance equal to the depth of said beam so as to be secured to the fore-and-aft stringer worked under said beam at side of hatches. The vertical stringer must equal the scantling of the deck beams, and the coaming plate must at least equal the thickness of the deck stringer which adjoins them, with an angle bar securing them to said stringer. (Rather an obsolete practice in view of present "Lake" methods.)

26. Deck Houses and Skylights.

1. Deck houses fitted on a weather deck over an engine or boiler opening must be of steel, of at least the same thickness as the upper half of watertight bulkheads. Stiffeners, equal to the reverse bars, are to be spaced not over 30 inches apart, with roof beams of same scantling located at every stiffener.

2. In hurricane-deck vessels, a steel trunk with a coaming plate is to be fitted between the hurricane and second

decks around the opening for the machinery. Said coaming plate at the second deck shall be similar to that for a weather-deck opening; that on the hurricane deck being of approved construction and height.

27. Wooden Decks and Ceilings.

1. The margin plank is to be of some approved hard wood.

2. All steel decks must be well painted before a wooden deck is laid thereon. When an oak deck is worked, felt or some other insulating material should be fitted between the steel plating and the wood decking.

3. A reduction of 1-2 inch from the tabulated thickness will be allowed in a wood deck which is laid on a steel deck extending the entire length of the vessel.

4. The main deck planking is to be fastened to the deck beams with galvanized screw bolts, with nuts on the underside; the heads of the bolts being bedded with oakum and white lead.

5. The deck planking of the hurricane, full poop, and enclosed bridge house decks may be 1-2 inch less in thickness than that of the main deck; but not to be less than 2 inches thick. It may be secured to the deck beams with wood screws.

6. The deck plating of a topgallant forecastle, raised quarter and fore deck, is to be similar in all respects as to thickness and fastenings to that of the main deck.

7. The hurricane deck of passenger vessels may be of the tongued and grooved type not less than 1 1-8 inches thick, being covered with canvas painted, so as to make it perfectly watertight. Said deck planking can be nailed to wooden battens fastened on top of the iron beams. When said iron beams are fitted on alternate frames a wooden beam should be worked between said iron beams. The cabin floors of yachts can be built in the same manner.

8. The lower deck of a hurricane-deck vessel is to be of the same thickness as that of the main deck; but it need not exceed 3 inches.

9. All parts of cargo spaces are to be close ceiled to the upper turn of the bilge, with battens worked berth and space above said point. In vessels engaged entirely in the coal, ore, or grain trade, said battens may be omitted.

10. A wooden ceiling, at least 2 1-2 inches thick, is to be worked over all inner-bottom plating, in such a way as to leave a drainage space between the steel plating and wooden ceiling.

28. Rudders.

The pintles are to be spaced from 4 1-2 to 5 feet apart. In vessels of less than 500 tonnage said pintles can be forged solid on the rudder frame, but for vessels of 500 or more tonnage they should be of hard steel let into the frame. The rudder frame should have a solid stay in way of the pintles. Side plates should be of the same thickness as the lower half of bulkhead plating, the space between the side plates being filled with wood. Rudders should be arranged so that they can be unshipped while the vessel is afloat. The pintles and the stock of a balanced rudder are to be 30 percent larger in sectional area than otherwise. Balanced rudders should have a thrust bearing at the main deck.

29. Windlass, Pumps, and Valves.

1. An iron windlass is to be fitted on all vessels requiring cables 3-4 inch in diameter or above. All other vessels should have a suitable capstan for handling the chains.

2. Sailing vessels under 200 tons are to have at least one hand bilge pump. When of 200 tons or above, two such pumps must be provided. Tug-boats are to have one hand pump for pumping the bilge. All of said pumps to be located on the upper deck.

3. Sluice valves, worked from the upper deck, are to be fitted to all watertight bulkheads, except to the forward collision and aft peak bulkheads, which are to be fitted with brass cocks.

4. A sounding pipe must be located in each watertight compartment, with a doubling plate to the shell plating under same. Two air pipes are also to be fitted to each inner-bottom compartment.

5. Brass cocks are to be fitted next to the shell plating on all lead pipes drawing from the sea.

6. Soil pipes are to be of iron up to the deep load water-line, with a foot valve riveted to the shell plating and caulked watertight.

7. All metal work should have three coats of good paint. The inside of the shell plating should be cemented to the turn of the bilge.

8. Means must be provided for the proper ventilation of all compartments not ventilated by skylights, side ports, etc. Storm shutters are to be fitted to all dead lights in cargo spaces.

30. Life Boats.

Tonnage.....	Under 100.	100 to 500.	500 to 1,000.	1,000 and above.
Number of boats....	1	2	3	4

One boat should be of sufficient size to transport the stream anchor.

31. Hurricane-Deck Vessels.

1. The depth for calculating the tonnage, and for obtaining the scantling number for frames, reverse bars, and floors, is to be taken to the deck next below the hurricane deck. The depth for regulating the ratio of depth to length and for calculating the equipment number is to be measured from the base line to half deck space between the hurricane deck and the deck next below said hurricane deck, at sides of the ship.

2. The hurricane-deck sheer strake should maintain its midship thickness throughout. The side plating between the main and hurricane-deck sheer strakes may be reduced by 2 1-2 pounds for 1-4 length at the ends, when its midship weight is 12 1-2 pounds or over.

3. Beams of the hurricane deck should be placed on alternate frames if the thickness of the deck plank will permit of caulking. When said deck is built of thin wood with a canvas cover a wooden beam must be worked between the iron beams that are worked on alternate frames.

32. Full Poops, Topgallant Forecastles, Inclosed Bridge Houses.

1. Deck beams, plate stringers, and tie plates are to be governed by the same rules as stated for hurricane-deck vessels.

2. The plating at the front of a full poop and inclosed bridge house is to equal the lower half of watertight bulkheads, with stiffening bars, equal to frames, spaced not over 30 inches apart. The coaming plate at the front of a full poop is to be 5 pounds heavier than said bulkhead plating, and shall stand at least 9 inches above the deck planking. The angle bar connecting said coaming plate to the deck plating is to stand at least 1-2 inch above the deck planking.

3. The depth for regulating the tonnage scantling for frames, reverse bars and floors, and for determining ratio of length to depth, is to be measured to the main deck at sides. When the length of either of the above structures, or of the combined length of any two, or of all of said structures exceeds 1-2 the vessel's length, the scantling tonnage increased by 1-8 of itself is to be the equipment number.

33. Raised Quarter and Raised Fore Decks.

1. The front of a raised quarter deck is to be of the same thickness as the outside shell plating in way of said quarter deck above the main-deck sheer strake, at the fore end of said structure. When said quarter deck stands more than 18 inches above the main deck its front must have stiffeners equal to the main frames, spaced not more than 30 inches apart. The angle bar connecting the bulkhead plating to the main-deck plating must stand above the deck planking so as to be caulked watertight. When no deck planking is worked said bar must equal the scantling of the main frames.

2. The after end of a raised fore deck is to be of same thickness as the plates in the upper half of a watertight bulkhead. Otherwise to be similar to the bulkhead at the front of a raised quarter deck.

3. Companionways must not be cut in front of a raised quarter deck; or in the aft ends of a raised fore deck, except when said bulkhead extends from the floors to the raised fore-deck beams.

34. Spacing of Rivets.

	DIAS.
1. Laps or seams of the shell plating	4
2. Laps or seams of all internal watertight work, chain fashion	4 1-2
Center to center of rows	2 1-2 to 3
3. Beam knees: to have at least 4 rivets to frames and beams	4 1-2 to 5
4. Bar keels; stems; stern frames; rudder plates, zigzag fashion	5
Center to center of rows	2 to 2 1-2
5. Frames to shell plating and floors; reverse bars to floors and frames; keelson and hold-stringer angle bars; all bulkhead, trunk casing, and centerboard-trunk stiffeners; and all similar non-watertight work	8

6. Treble-riveted butts to have three complete rows.

7. Shell clips to have four rivets in each flange.

8. No rivet is to be nearer the seam, butt, or edge of a plate or angle bar than its own diameter.

TABLE 12.—AMERICAN BUREAU OF SHIPPING. WIDTH OF BUTT LAPS AND BUTT STRAPS. SPACING OF RIVETS. FOR SOUND, LAKE, BAY AND RIVER SERVICE AND YACHTS AND TUGS.

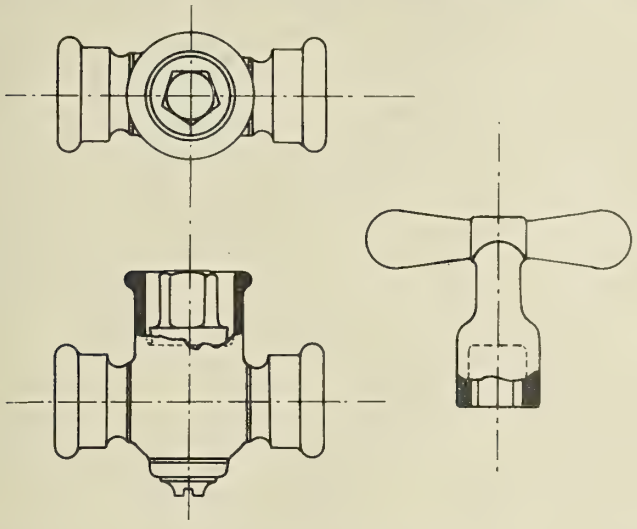
	Weight of Plate. Lbs.	Diametral Pitch.	4 to 6	8	10	12 to 16	18 to 22	24 to 28	30 to 34	36
	Diameter of Rivet. Inches.		3/8	7/16	1/2	5/8	3/4	7/8	1	1 1/8
Widths of:	Single riveted butt lap	....	1 3/8	1 5/8	1 7/8	2 1/4	2 5/8	....	....	....
	Double " " "	....	2 1/8	2 1/2	2 7/8	3 3/4	4 3/8	5 1/8	5 3/4	6 1/4
	Treble " " "	....	....	....	....	....	....	....	....	....
	Single " " strap	7	2 5/8	3 1/16	3 1/2	4 3/8	5 1/4	6 1/8	7	7 7/8
	Double " " "	....	4 1/4	5 1/8	5 3/4	7 1/4	8 1/2	10 1/4	11 1/2	13
	Treble " " "	....	6 1/2	7 5/8	8 1/2	10 5/8	12 3/4	15	17	18 1/2
	Single " seam lap	....	....	....	....	....	....	....	....	....
	Double " " "	....	....	....	....	....	....	....	....	....
	Single " edge strip	7 to 8	2 5/8	3	3 1/2	4 3/8	5 1/4	6 1/8	7	7 7/8
	Double " " "	11 1/2 to 12 1/2	....		....	7 1/4	8 3/4	10 1/8	11 1/2	13
Spacing of rivets.	1.—Center of rivet from edge of plate	1 1/2	9/16	11/16	3/4	15/16	1 1/8	1 5/16	1 1/2	1 11/16
	2.—Zigzag riveting: center to center of rows	2 to 2 1/2	....	....	....	....	....	....	-	....
	3.—Chain " " " " " "	2 1/2 to 3	....	....	....	....	....	....	....	....
	4.—Seams or laps of the shell plating	4	1 1/4	1 3/4	2	2 1/2	3	3 1/2	4	4 1/2
	5.— " " " " all inside watertight work	4 1/2	1 11/16	2	2 1/4	2 13/16	3 3/8	3 15/16	4 1/2	5 1/16
	6.—Beam knees	4 1/2 to 5	....	....	....	....	....	....	....	....
	7.—Bar keels; stems; stern posts; rudder plates	5	1 7/8	2 3/16	2 1/2	3 1/8	3 3/4	4 3/8	5	5 5/8
	8.—Ordinary frames to the shell and floors; reverse bars; keelson and stringer bars; bulkhead stiffeners: all non-watertight work	8	3	3 1/2	4	5	6	7	8	9

A Bennington Suggestion.

Editor Marine Engineering:

As it appears from the report of the board of investigation the probable primary cause of the lamentable accident to the U. S. gunboat *Bennington* was the "shutting off" of the steam gauge on the boiler, the idea suggests itself to me that it would be advisable to provide a means of preventing a repetition of such a deplorable mistake. While it is quite true that an accident of this kind might not happen again in a century, the fact remains that it *has* occurred, and any means of preventing another accident of the kind would add still further to the numerous devices for the safety of boilers, which are necessitated by that never-to-be-neglected factor, the "personal equation."

It has occurred to me that, in lieu of the ordinarily adopted handled cock or valve, it would be well to use a cock similar to the one shown in the accompanying sketch, wherein it will be



seen at a glance that it cannot possibly be closed without a special handle or key. This handle or key should be kept in the engine-room desk, or other place where it can be accessible only to the engineer of the watch. It is extremely seldom that it becomes necessary to shut off a steam gauge from the boiler, and when an occasion of this kind does arise, the engineer in charge should be the one to do it, and time can always be taken to go into the engine room for the key. There is nothing original about such a device, as will be mutely testified by the thousands of gas and water plugs, steam radiator valves, etc., in common use; but had such a cock been fitted to the steam-gauge pipe on the *Bennington*, the water tender, or whoever it was who attempted to close the air cock and succeeded in shutting off the steam gauge, would have been thwarted in ignorantly making a mistake which led up to the most frightful accident to steam machinery that has occurred in modern times. Attention to details which may appear trifling is, as everyone will admit, the secret of success in the safe operation of machinery of all types, and the foregoing suggestion is offered to the engineering profession for its consideration in that line.

C. A. McAllister,
Chief Engineer, U. S. R. C. S., Engineer-in-Chief.

The Bureau of Navigation of the Department of Commerce and Labor reports that there were built in the United States during the month of July 116 vessels of an aggregate of 12,706 gross tons. As this list included one ship of 6,363 tons, or more than half the total, it will be seen that the remainder consisted of 115 ships of an aggregate of 6,343 gross tons, or an average of *55 tons* each.

The figures for August are a little more encouraging, there having been 107 vessels of an aggregate of 44,028 gross tons, of which five, each of more than 4,000 tons, accounted for a total of 29,221 tons. This leaves a balance of 102 vessels with an aggregate of 14,807 tons, or an average of *145 tons* each.

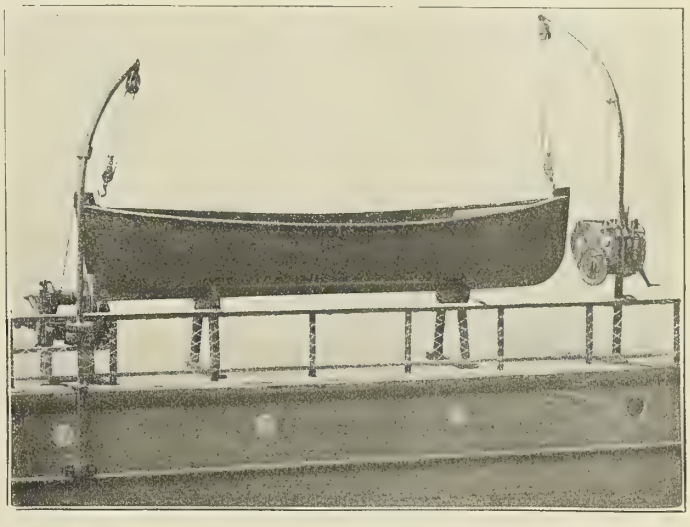
The thirteenth general meeting of the Society of Naval Architects and Marine Engineers will take place in New York city at 10 A.M. Thursday, November 16, 1905. The meetings will be held in the Auditorium at No. 12 West Thirty-first street, the sessions continuing through Thursday and Friday, November 16 and 17. The banquet following the sessions of the meeting will be held at Delmonico's, Fifth avenue and Forty-fourth street, on Friday evening, November 17, at 7 o'clock. The programme giving the list of papers to be read at the meeting will be ready about October 15.

Mr. H. F. J. Porter has just opened an office in the Metropolitan Building, 1 Madison avenue, New York, N. Y., for a consulting industrial engineering practice in connection with installing modern methods of organization and management. Mr. Porter has had a valuable experience extending through more than a quarter of a century of work of great variety in character and scope, and should make a splendid success at his new undertaking.

ENGINEERING SPECIALTIES.

Automatic Safety Life-Boat Handler.

The Automatic Safety Life-Boat Handler is the result of some ten years' work and experience of a practical sailor, Captain Frank W. Irvine, late of the Ward Line. It is a noticeable fact that, although many and important improvements have been made in the construction of vessels and steamships and their equipments during recent years, the facilities for handling life-boats, so important in saving lives, have improved very little during the last century. The hoisting, swinging out, and lowering away of life-boats by this device can be done in a half to one minute in all kinds of weather by only two men. Guys are not needed. The old method of doing this work requires ten men from seven to ten minutes.



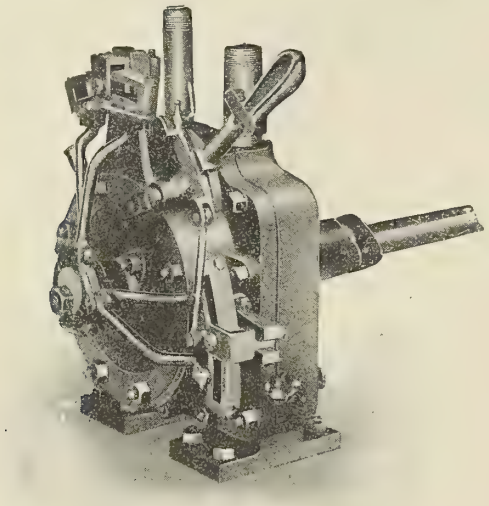
When life-boats are most needed as a rule electricity or steam cannot be depended upon. This gear, which is worked by hand, can be attached to any davit now in use. After the boat is hoisted the davits swing out automatically, no pulling or pushing is required, and the lowering away of the boat is entirely in the safe control of the two men in charge of the gears. The gear may be used on life-rafts as well as on life-boats. The usual confusion and delay is avoided and the crew have ample time for careful inspection of the boats and the care of the passengers. The two men necessary to handle the gear need not be sailors; simple instructions are attached so that any two passengers, in an emergency, can lower the boat.

The two machines required for each life-boat occupy a deck space beside each davit of 14 inches by 24 inches. The only additional deck space used in handling the boats is that required for a

man at each machine, who works the machine and the falls. Wire rope is used for the falls, as a further protection against fire. In cold weather ropes are generally frozen and often full of kinks and turns, which greatly handicap the work of lowering the boat Wire will last much longer and will, in the course of time, save the cost of gear, as it is necessary to renew rope at least once a year. The device is being handled by the Barcus-Hallam Company, 111 Fifth avenue, New York.

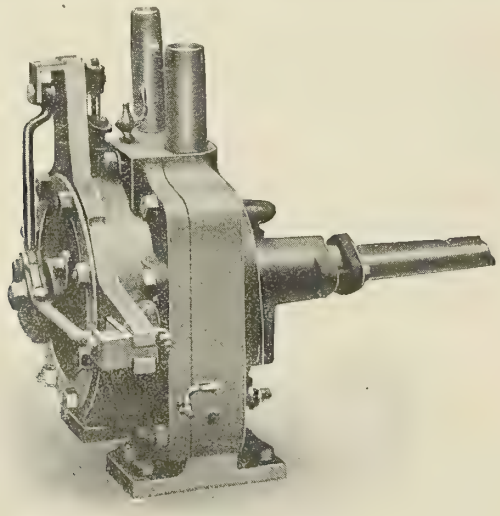
Werner Fluid Motor.

Strictly speaking, this motor consists of two distinct reciprocating engines so placed that the strokes of their pistons are at right angles to one another. There are two cylinders, one within



REVERSING MOTOR.

the other; two pistons, also one within the other; a crank pin and a crank shaft in common for both pistons; two valves; two valve rods; two eccentrics; two links, etc. The outer casing constitutes the outer cylinder for the outer piston, and the latter constitutes the inner cylinder for the inner piston. The outer piston only reciprocates, while the inner piston travels in a circular path with



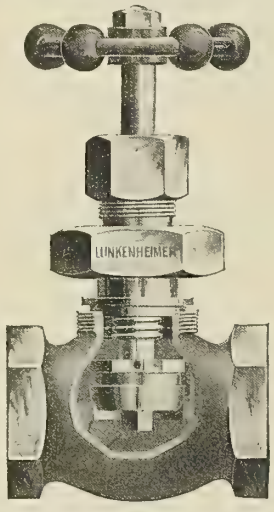
NON-REVERSING MOTOR.

reference to the main body of the machine. The inner piston reciprocates only with reference to the outer piston and its to-and-fro motion is on line perpendicular to the upper and lower bearing surfaces of the outer piston and not perpendicular to the line of movement of the outer one.

Steam, air, or whatever the motive fluid may be, is admitted successively to both ends of each piston, thus giving the motor shaft four successive impulses for every revolution. The valve gear is practically the same as is used on the locomotive, there being two double D-valves in a common valve chest, two valve rods, two links, four connecting rods for the valves, and only two eccentrics instead of four. Two of these are dispensed with since the two valves lie in the same plane and in the same chest. In this gear the valves lie on the opposite side of the link arc as is generally found in the Stevenson valve gear, but the characteristic functions are identical in every respect with that of the latter. There is nothing new in this motor—only new applications of principles already known.

Improved Regrinding Valves.

A number of desirable improvements on the Lunkenheimer (Cincinnati, O.) regrinding valves have been made, resulting in an increase in the weight of the valve, as an additional precaution against rough handling while attaching. The medium pattern valves are guaranteed to stand a working pressure of 200 pounds, and the extra heavy pattern up to 300 pounds per square inch. The shape of the valve has also been altered, increasing the area through the valve and permitting an unobstructed passage through same, the area being now more than equal to that of the connecting pipe. To insure a perfect and strong joint between the pipe and valve, the pipe threads have been made considerably longer, thereby obviating the danger of stripping threads. All sizes of the valves now have lock nuts on the hand wheels, which facilitates the taking apart and assembling of the valve for repairs, etc.



Referring to the sectional illustration, it will be noticed that the hub which carries the operating stem is secured to the body by a union connection, which, in turn, screws over the shell of the valve body. By means of this construction it is impossible for the hub and the body to become corroded together, as the thread which holds the union ring to the body is protected at all times from the action of the steam, the joint being made between the flange on the hub and the neck of the body.

To regrind the valve, the bonnet ring is unscrewed, and the trimmings are removed from the body. A wire or nail is placed through the lock nut and stem, a little powdered sand, or glass and soap or oil is placed on the disc, and the trimmings are again placed on the valve and reground.

TECHNICAL PUBLICATIONS.

The Naval Pocketbook. By Sir W. Laird Clowes. Pages 947 + xxxiii. London: W. Thacker & Company, 1905. Price 7s. 6d. net.

What will probably prove to be the last of this series of very useful books is the 1905 edition of "The Naval Pocketbook," which

is now in its tenth year. Our reason for assuming the immediate cessation of this series is the death of Sir W. Laird Clowes, which occurred a few weeks ago. It is much to be regretted that there is little hope for a continuation of the series, because of the fact that an immense amount of valuable information regarding warships of all the world has been collected in the little volume, which has been corrected to March 31, with addenda bringing the total up to June 9, 1905, and including such recent items as the results of the naval battle of Tsushima. The contents include a complete list of every battleship, cruiser, gunboat, torpedo boat and destroyer and submarine in all the navies of the world; notes on torpedoes; comparative fighting strength and navy expenditures of the powers; list of dry-docks; steam trials of ships; plans of a large number of the most important ships afloat; a complete index of ships by names; and the usual tables for conversions of measures between the metric and English systems.

One of the most interesting features of the book is a table contained in the preface, giving a comparative summary of the fighting fleets of the world in ships built and building.

We have received from L. Smit & Zoon, of Kinderdijk, Holland, a very interesting little book bound in cloth and containing 137 pages and 89 illustrations. While this book is primarily a house organ, and is devoted almost exclusively to the exploitation of the products of the firm which issued it, yet the illustrations and the general character of the descriptions offered render it of especial interest to marine engineers in general, and particularly to those concerned with the art of dredging, either as a designing engineer or as an operating engineer. The work is printed in six languages, which are given in the book in the following order: Dutch, French, German, English, Russian, and Spanish. A very novel index is provided in which the page numbers are set opposite the flags of the countries in which the six several languages are spoken, there being a total of 15 flags represented, this necessitating the printing of that page in five colors. The half-tones scattered throughout the work seem to be about equally divided between full-page cuts and small capitals and tail pieces, while a number of line drawings have been used to illustrate special appliances.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the Editor in this column. All communications must bear the name and address of the writer.

Q. 298.—Kindly answer the following questions for me relative to an engine 5½-inch bore, 6-inch stroke, running 450 R.P.M., a 2-cycle gasoline motor.

(*a*) What percent of the piston displacement should the clearance be to give proper compression?

(*b*) Give the proper size and area of inlet and exhaust ports.

(*c*) Is a plunger pump more satisfactory than a rotary and why?

(*d*) What is generally taken as an average M.E.P. of this style engine?

(*e*) Is the reversing engine as manufactured by some makers reliable and sure, so that no reversing gear is necessary?

(*f*) Give me the names of some reliable marine engine reversing gears?

(*g*) Is there a patent on a gas engine piston for 2-cycle engines having a port in the side near the head to correspond with the inlet port in the cylinder, and if so what is the patent number? F. G. K.

A.—(*a*) The percentage of piston displacement taken for clearance volume should be from 30 to 35, varying with the richness of the gas. The poorer the average mixture the higher the compression should be and hence the smaller the clearance volume. This proportion will insure a pressure at the end of compression of 80 to 90 pounds, and an initial pressure after combustion of 300 or upward. Too small a clearance volume will mean too high a compression pressure, a possibility of premature ignition due to the heat developed, and greater irregularity in the motion of the engine. Too large a clearance volume, on the other hand, will mean too low a terminal pressure, liability of incomplete combustion, and loss of effect. In the case of the engine you mention, with stroke of 6 inches, the clearance length should be about 2 inches to realize good average results.

(*b*) The inlet and exhaust ports are sometimes made about equal in area, but the latter should be preferably some 30 to 40 percent the larger of the two. The size is chiefly determined by the displacement volume of the piston and the number of revolutions per minute. The problem is simply to provide an inlet passage of such area that with the slight elevation of pressure in the crank case a cylinder-full of vapor will readily enter in the small time allowed near the end of the stroke, and similarly such an area of exhaust port that the cylinder-full of spent gases will escape with rapid drop of pressure in the small time likewise allowed at the end of the stroke. Experience indicates that to fulfil these conditions the inlet port should have an area substantially as given by the formula:

$$a = \frac{NV}{36{,}000}$$

where a = area of inlet port in square inches.
N = revolutions per minute.
V = displacement volume of piston in cubic inches.

For the exhaust area some 30 to 40 percent in excess should be allowed.

For the engine you mention the inlet passage should have an area not less than 1¾ to 2 square inches, while the exhaust area should be about 2¼ square inches.

(*c*) The satisfactory operation of the circulating pump will depend more upon its actual design and construction in detail than upon the type—whether plunger or rotary. Satisfactory pumps of both types are in common use.

(*d*) The mean effective pressure of such an engine should be about 90 or 100 pounds. This results as a combination of clearance volume and compression line, quality of mixture and completeness of combustion, and area of inlet and exhaust passages.

(*e*) The only type of reversing engine which can be recommended as satisfactory (except perhaps in the very smallest sizes) is that in which a multi-cylinder engine is provided with an arrangement for moving the valve gear and sparking mechanism into either ahead or backing gear, and in which, in addition, a compressed air chamber is provided for giving the necessary initial impulse in starting. Further reference to this point will be found in Marine Engineering for September in the article on "Motor Boats." With such an arrangement and a little skill in manipulation, the handling of a gas engine becomes very similar to that of a steam engine provided with the usual type of reverse gear.

(*f*) We cannot recommend by name specific types of reverse gear in these columns. We should advise you to send for circulars from the various advertisers in this journal and to determine for yourself on the plain merits of the various arrangements offered.

(*g*) We cannot search patent records for items such as that asked for. Any patent attorney will be able to aid you in regard to such matters.

Q. 302.—I always put the zincs in the boiler in baskets and usually fasten them with bolts, but I have about 15 baskets without the bolts to fasten the zincs on. It is claimed that the zincs must be fastened so as to form a metallic contact. From my experience with the 15 zincs taken out without bolts, they are just as much eaten away as the others, so I claim that they don't need the bolts. Now I want to know if it makes much difference whether they are fastened with bolts or not. W. B. M.

A.—Zinc slabs should be provided in dimensions of about one-half square foot with a thickness of one inch for each 25 or 30 indicated horsepower, and should be well distributed throughout the boiler. They are often suspended from the stays, and care should be taken that the zinc, the supporting strap, and the stay touch one another with a good bright metallic contact in order to insure the permanency of the galvanic connection. When in baskets, if they make good contact without the bolts, that is sufficient, but the bolt makes the contact certain.

The use of zinc in this connection arose from the accidental leaving of a bar of zinc in a boiler which had just been cleaned.

and which at the end of the voyage was found to be entirely free from scale, while the zinc had disappeared.

Q. 303.—Would a fleet of war vessels on the Hudson River increase the weight on the roof of the Pennsylvania tunnel just beneath them or further down stream, and to what proportion of their weight? Over how much river bed would the weight of these vessels be spread? K. S.

A.—A fleet of war ships on the Hudson River immediately above the Pennsylvania tunnel, or at any other position in the neighborhood, would have no effect whatever upon the load pressing upon the roof of the tunnel. This is readily explained by the fact that each ship displaces or pushes aside an amount of water equal in weight to the weight of the ship, and that, therefore, the general level of the water in the river would not be increased by the presence of the warships therein, and so the pressure upon the tunnel, which will depend directly upon the depth of the water over the tunnel, would not be increased by the ships.

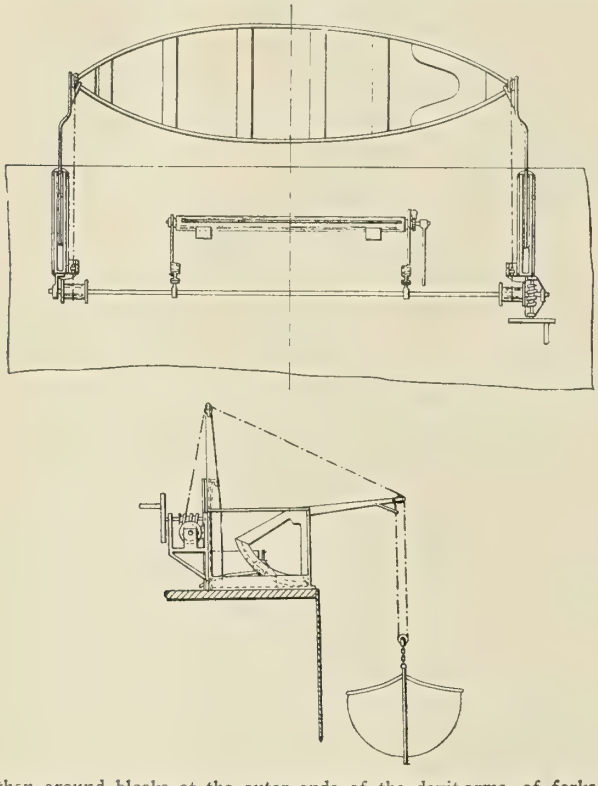
Q. 304.—At a recent examination a question was given as follows: A boiler 72 inches diameter with 80 pounds pressure; what is the stress per square inch on the longitudinal seam? Is this sufficient data for this problem, or should the length and the thickness of shell have been given? J. M.

A.—The stress per square inch on the longitudinal seam of a cylindrical boiler is expressed by the formula $S = pd \div 2t$, where p represents the boiler pressure in pounds per square inch, d represents diameter of the boiler in inches, and t the thickness of the shell in inches. It will be seen that in order to obtain the stress in the example in question it will be necessary to know the thickness of the plate. Assuming the thickness to have been ½ inch, the stress figures out as 5,760 pounds per square inch.

SELECTED MARINE PATENTS.

795,937. APPARATUS FOR RAPID LAUNCHING OF LIFE-BOATS. GEORGE CHRISTIAN SCHMIDT, GOTEBORG, SWEDEN.

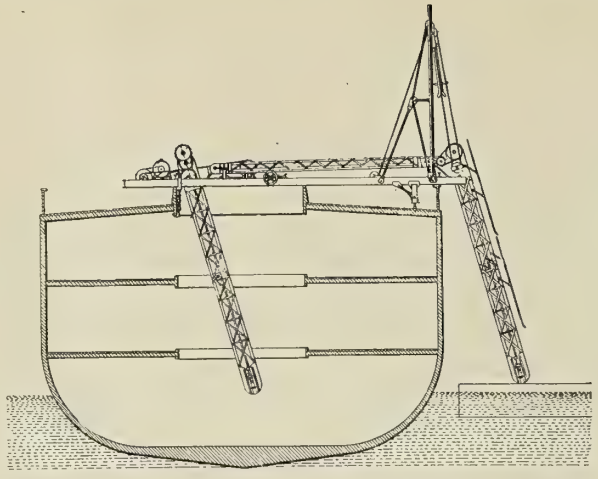
Claim.—3. In a device for rapidly launching life-boats, the combination with davit-arms adapted to oscillate in vertical planes, and provided at their outer ends with hoisting-ropes secured thereto, each passing around a block which is secured to and can be readily detached from the life-boat,



and then around blocks at the outer ends of the davit-arms, of forks projecting from the outer ends of the davit-arms and forming with the said davit-arms crutches, and of posts provided at their tops with pulleys, and two drums arranged on a rotatable shaft in common to both, for the purpose specified. Seven claims.

796,406. CARGO-CONVEYER. CHARLES H. ANDERSON, CHICAGO, ILL.

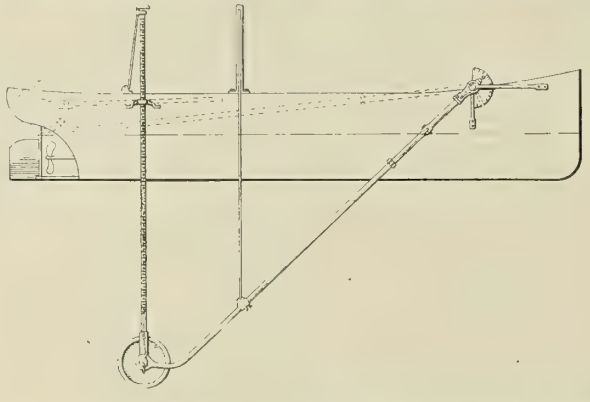
Abstract.—This invention relates to a cargo-conveyer devised more especially for loading packages of wood-pulp or the like into ships from wharves or lighters. In practice the conveyer mechanism when not in use is stored



upon a barge or dock. The barge or dock is provided with a derrick suitable for handing and placing the conveyer sections or members in position upon a ship and for removing the conveyer therefrom when it has done its work. The principal features of novelty reside in a portable support for the conveyer-sections, with means for leveling it upon the deck of a ship, the support carrying the electric motor or other power mechanism for operating the package-conveyers and having means in combination for holding the conveyer-sections in mutually co-operating relation during the rocking movement, etc., of the ship. Fifteen claims.

796,504. SOUNDING DEVICE. SWEPSON EARLE, WASHINGTON, D. C.

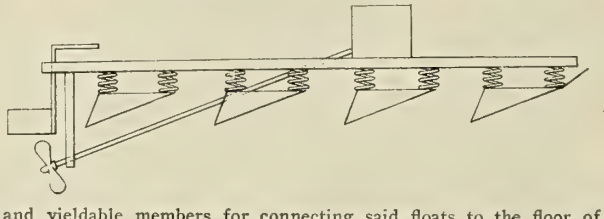
Claim.—2. A sounding device comprising a contacting member, an inclined pivoted arm carrying said member and having a rotary sleeve-joint, an in-



dicating device, a signal, an arm connecting signal and the contacting member, both indicating device and signal being actuated by the movement of the contacting member. Twelve claims.

796,846. RAFT OR OTHER CRAFT. CHARLES A. DE LAMBERT, VERSAILLES, FRANCE.

Claim.—In a raft or other flat-bottom craft having motive power, a plurality of floats of prismatic form with their axes perpendicular to the longitudinal axes of the raft, and lower faces oblique to the bottom of the



raft, and yieldable members for connecting said floats to the floor of the raft. One claim.

797,146. SECTIONAL CENTERBOARD FOR VESSELS. VIGGO L. OGIDINSSEN, SAN FRANCISCO, CAL.

Claim.—7. The combination with a vessel, of guides on the bottom thereof and running fore-and-aft, a centerboard movable in said guides, and means operatable on the vessel to shift the position of the centerboard in said guides or to remove it entirely from the bottom of the vessel or to replace it in position there-beneath. Eleven claims.